The Association
of
Engineering and Shipbuilding Draughtsmen.

STEAM LOCOMOTIVE DESIGN.

PART II.

By R. H. MANN.

Published by The Association of Engineering and Shipbuilding Draughtsmen
Onslow Hall, Little Green, Richmond, Surrey.

SESSION 1957-58.

Price 6/-

Printed by Milne, Tannahill & Methven, Ltd. (T.U.), 12-14 Mill Street, Perth.

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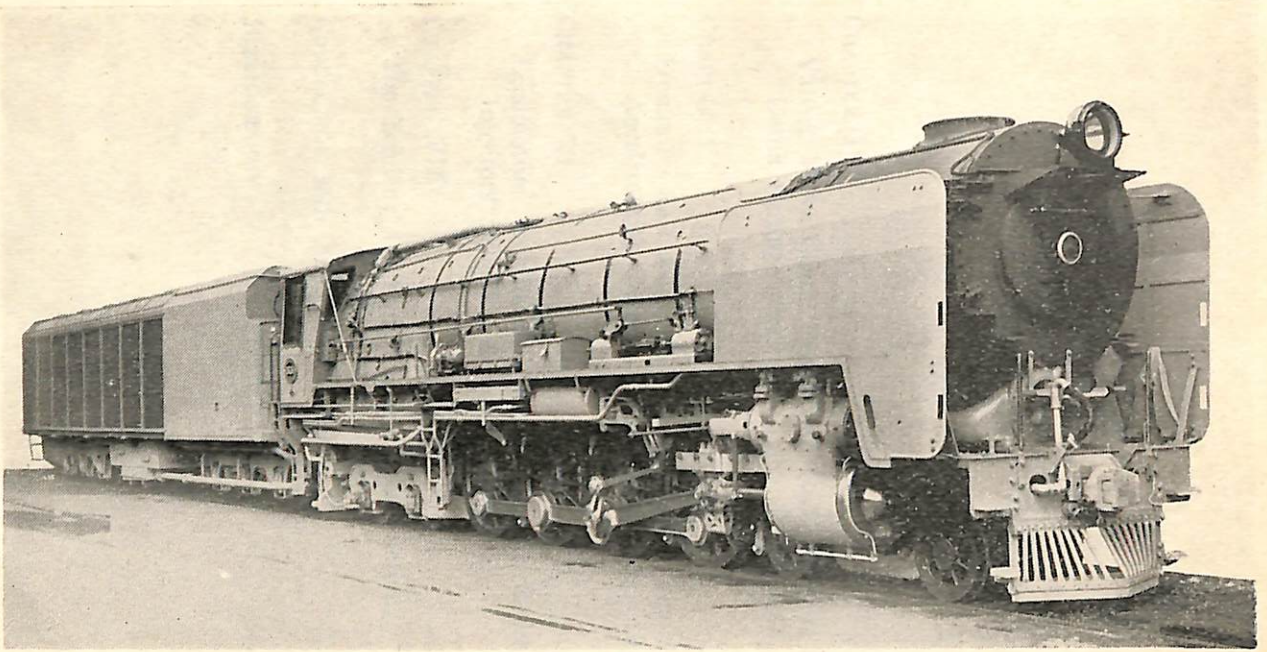
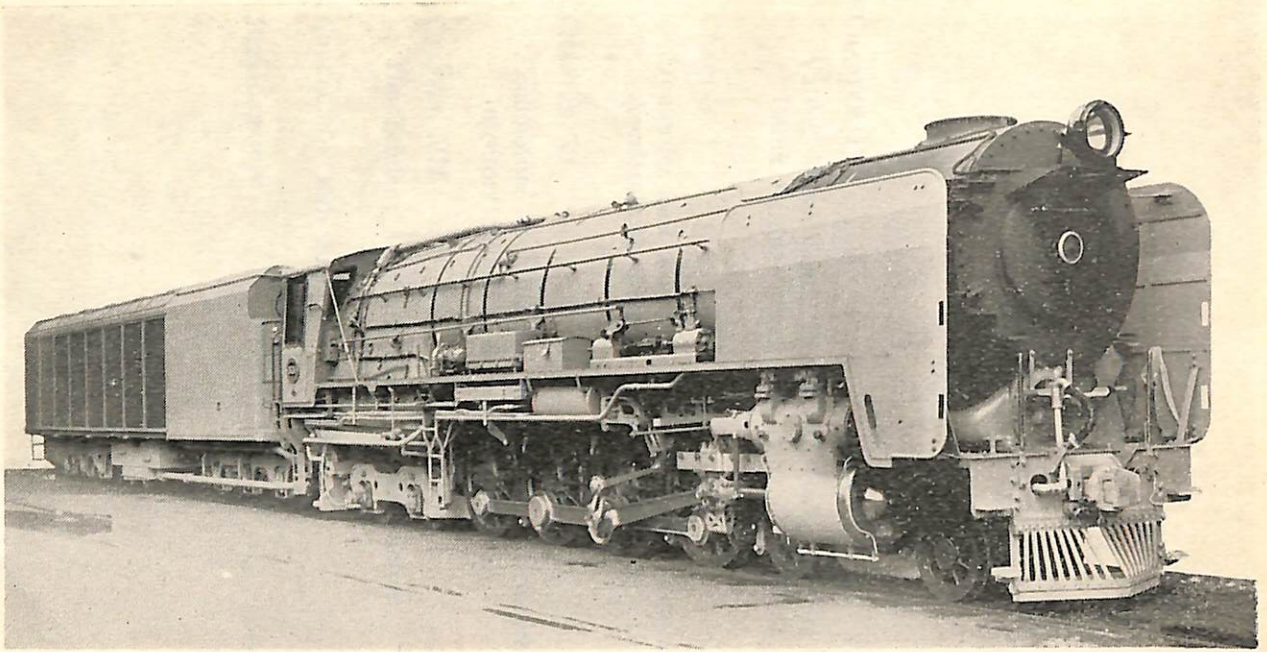


Frontispiece (Part II). *[Courtesy N.B. Locomotive Co., Ltd.*

South African Railways, "25 Class," 4-8-4, Condensing Locomotives (3′ 6″ Guage).

Cylinders (2), ... 24″ × 28″. Coupled Wheels, ... 5′ 0″ dia. Boiler Pressure, ... 225 lbs./sq. in.
Tractive Effort at 75%, 45,360 lb. Weight Engine, ... 121 tons. Total Engine and Tender, 230.5 tons.
One of 90 locomotives built by N.B. Loco. Co., Ltd., for the South African Railways, a fine example of the locomotive builders' product for colonial and dominion railways. Fitted with D. & M. F.S.A. valveless lubricator and The Superheater Co.'s combined multiple valve regulator and Superheater.



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CONTENTS.

PART II.

Page

CHAPTER IV 7

Valves, Valve Gears, Piston and Rods, Motion Details, Crank Arrangements, Reversing Mechanism.

CHAPTER V 46

Dynamics and Associated Parts, Piston Velocity and Acceleration, Torque, Wheel Balancing, Braking.

CHAPTER VI. 70

Locomotive Auxiliaries, Fittings and Refinements, Injectors, Ejectors, Lubricators, etc.

CHAPTER VII 87

The Tender: Its Function and Construction.

CONCLUSION AND ACKNOWLEDGMENTS 104

BIBLIOGRAPHY 113

STEAM LOCOMOTIVE DESIGN.

PART II.

CHAPTER IV.

VALVES, VALVE GEARS AND MOTION.

It is well known that the steam locomotive can haul continuously heavy loads, at considerable speed over long distances, but were it not for a suitable and well-designed valve gear it would neither do this, nor move itself under its own steam no matter how big and efficient its boiler, nor whatever its evaporation and working pressure. An investigation as to how this is accomplished must now be undertaken.

(1) The Function of a Valve.

Is to effect correct steam distribution to each end of the cylinder alternately, which function entails (i) Steam Admission ; (ii) Steam Cut-off ; (iii) Steam Release, and (iv) Compression and Pre-Admission. For present purposes it will be assumed that the reader possesses a basic understanding of the well-known mechanism of slide valve and eccentric whereby steam distribution is effected in stationary engines, for there exists a vast bibliography on this subject. Its application to the steam locomotive will be the concern of this investigation.

Considering the cycle of events in the cylinder it is usual to assume the crank to be on dead centre, *i.e.*, piston rod, connecting rod and crank, to be in one straight line. To move the piston steam must be admitted to the cylinder, in fact steam must be admitted to the cylinder some 4° or 5° *before* the crank has reached dead centre. Thus *Admission* occurs before the end of the preceding stroke and continues until some such time as the valve cuts off the steam supply. To effect this the valve is given "Lead," *i.e.*, the amount of port[1] opening to steam at commencement of stroke, and is about $\frac{1}{16}$ in. to $\frac{1}{4}$ in. in locomotive design.

It would be wasteful to admit steam for the entire duration of the stroke, besides exceeding the ability of the boiler to supply it, so at some point the valve after having reversed its direction of travel, and dependent upon a great many factors commences to "cut off" by closing the cylinder port to steam. Thus "cut off" occurs during the initial stroke of the piston, and consequently

[1]Port is the term given to the openings in the cylinder and valve chest, of the interconnecting steam passage.

the steam fills the cylinder by expansion pushing the piston ahead of it, with a consequent fall in pressure. The expansion period should be as long as is consistent with correct release.

The point of "*Release*" should occur at about 85%-90% of the piston stroke, and is the instant at which the valve has opened the cylinder port to exhaust. The piston now reaches (the other) dead centre and reverses its direction of travel and pushes out of the cylinder ahead of it the steam which propelled it on its initial stroke, until some instant when the valve, having meanwhile reversed its direction of motion (*i.e.*, it is now moving in the same direction as it was when the cycle of events commenced) cuts off the cylinder port to exhaust (about 80% of the return piston stroke).

The steam remaining within the cylinder for cushioning purposes[2] becomes compressed and rises rapidly in pressure, which pressure should never be allowed to exceed the steam chest pressure, and is finally joined by a supply of fresh steam at some point about 85%-90% of the return stroke of the piston.

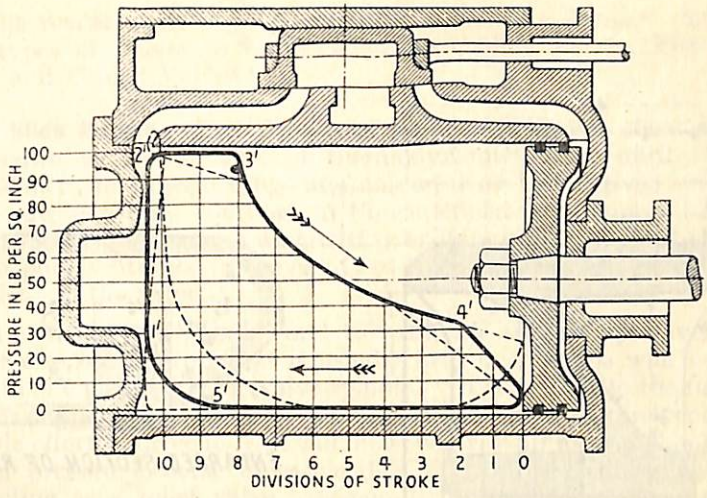
This point is known as "Pre-Admission" and is due to the action of the valve whereby boiler steam is admitted to the cylinder before completion of the return stroke ; so that pressure can be built up in the cylinder equal to the pressure in the steam chest, at or immediately after the commencement of the outward stroke of the piston. This completes the cycle of events within the cylinder itself, it being realised of course that a corresponding cycle of events is occurring at the other end of the cylinder on the reverse side of the piston 180° out of phase with the events just described.

The working of steam expansively is made possible by giving the valve "lap," *i.e.*, the amount the valve head (piston) or face (slide) overlaps the cylinder port on the steam side when placed centrally above it. To obtain the advantages of the "lap" valve it is imperative that its travel should be varied by very small amounts successively from maximum to minimum in relation to the constant throw of the eccentric driving it. This is best achieved by the interposing of a variable valve gear between the valve and eccentric, whose function is to drive the valve in such a manner that the maximum use is made of the expansive properties of the steam along with the minimum steam consumption possible under the conditions of working, in the process.

(2) Types of Valve Employed in Locomotives.

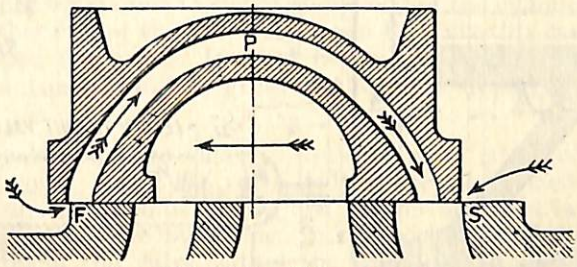
So far no mention has been made of what form the valve shall take, and so it is necessary to consider some of the valves in general use, though there exist many variations of these, and for the more

[2]Assists in retarding the piston, and provides a cushion for it to come to rest upon thus minimising inertia forces in the connecting rods, knocking and pounding in the axle boxes.

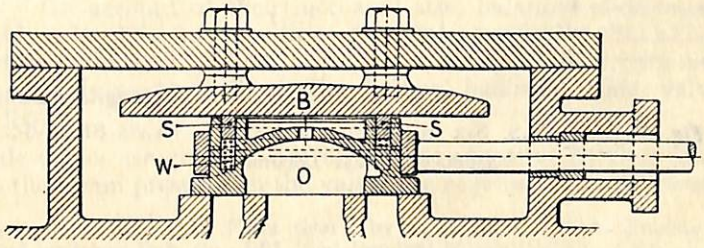


[*Courtesy Railway Gazette.*

Fig. 47—Indicator Diagram Superimposed on Locomotive Cylinder Showing Action of Steam.



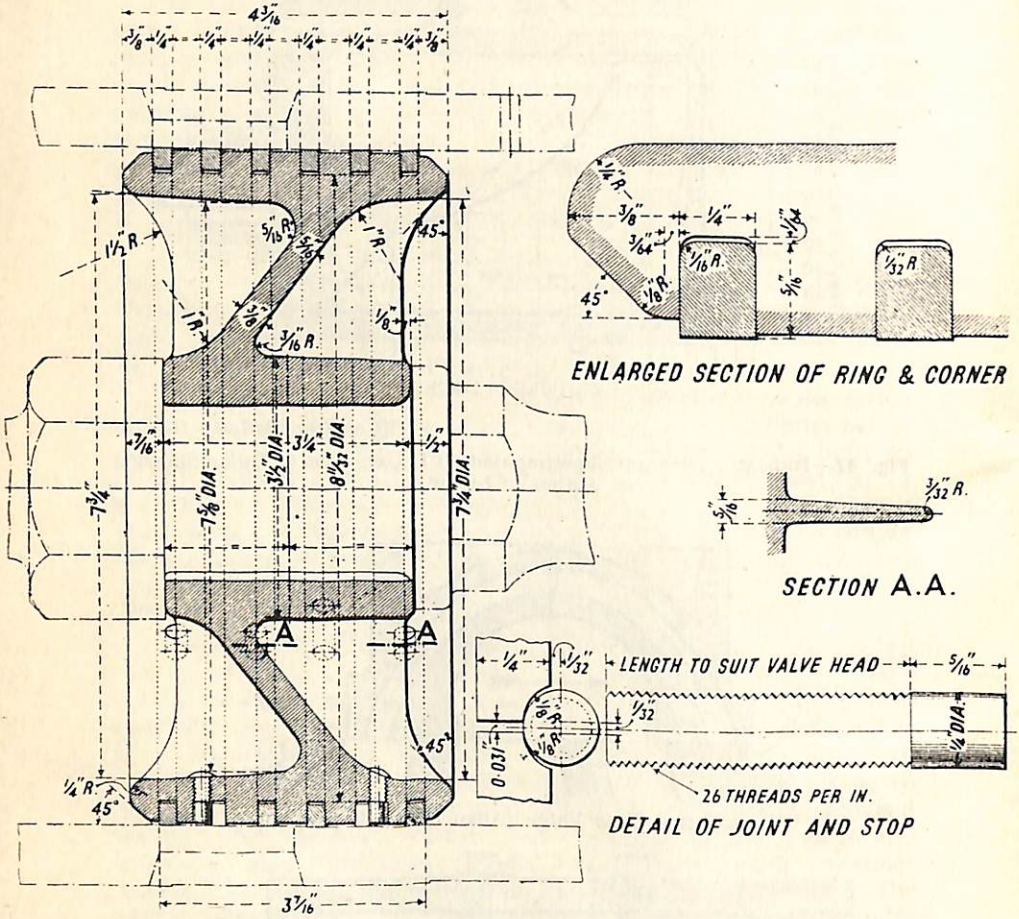
Locomotive Slide Valve (Allan or Trick Pattern).



Richardson Balanced Slide Valve for Locomotives.

[*Courtesy Railway Gazette.*

Fig. 48 (a)—Locomotive Slide Valves.



[*Courtesy Railway Gazette.*

Fig. 48 (b)—L.M.S. Six Ring Piston Valve. Class 8P. 4-6-2. "Duchess" Class Locomotive.

thorough reader a useful bibliographical reference is given.[3] The three types of valves to be considered are Slide Valves, Piston Valves and Poppet Valves.[4]

(i) **Slide Valves.**—The ubiquitous slide or 'D' valve must be well known to all students of thermodynamics, and, until the establishment of superheating, was universal on locomotives; its use is now generally confined to "unsuperheated" engines. In its usual form the steam is admitted over its outer edges, and the exhaust port is situated in the cavity of the valve chest underneath the "dish" of the valve.

It must be given "lead," and to work the steam expansively it must also have "lap." It is usually held in a buckle which is forged in one piece with the valve spindle. It is subject to the full pressure of the steam and if of the unbalanced type, requires considerable effort to drive it. It will, however, rise off its seat should cylinder pressure become excessive due to entrapped condensate, thus acting as a relief valve. Generally its events are sluggish and throttling of steam occurs. To alleviate this characteristic, modifications have been introduced whereby the "lead" is increased, the Allan or trick valve being the best known in which a by-passing cavity is cored out of the valve from front to rear in such a manner that when open to steam at one end of the cylinder, steam from the other end of the valve chest can flow via this cavity into the port open to steam. It must be realised with trick valves that the actual "lead" is greater than the apparent "lead." (Fig. 48a).

In the balanced slide valve, a pressure plate is bolted to the valve chest cover, whilst the valve body is deeply recessed to take cast iron strips arranged in an "Oxford" Frame and kept in contact with the pressure plate by springs in the bottom of the recesses. Thus the top of the valve within the frame is not under direct pressure, and the valve is subjected only to pressure of the exhaust steam. On account of their increased size, balanced slide valves are seldom used in any position other than vertically above their respective cylinders. There also exist combinations of trick and balanced valves, as well as the hollow balanced slide valve. (Fig. 48a).

Slide valves are given to heating due to excessive friction, itself due to the steam pressure on the valve, the easy incursion of foreign

[3]*Locomotive Valves and Valve Gears*, by C. S. Lake and A. Reidinger, published by Loco. Pub. Co., Ltd., and Percival Marshall & Co., Ltd.

[4]Sleeve valves were fitted experimentally to one of the "Brighton" L.B.S.C.R. Atlantics, by O.V. Bulleid, prior to their incorporation in the ill-fated "Leader" class tank locomotives scrapped by British Railways upon nationalisation before being placed in active service.

matter, and the difficulty of lubricating effectively the large area of valve face.

(ii) **Piston Valves** are in effect two pistons mounted on a common rod, whose faces and contacting seats (*i.e.*, the valve liner) are completely cylindrical. The piston valve is perfectly balanced with regard to steam pressure, and the only friction to be overcome is that due to the small side pressure required for steam tightness, in all 12½%-20% of that of the equivalent slide valve. A great variety of designs exist or have existed including the trick[5] piston valve, Smith, Robinson, Schmidt, American Semi-Plug, the "L" Ring Valve, Willoteaux and Troffinoff,[6] all designed with some particular end in view, so for present purposes only the general type of valve will be considered.

In this type of valve the heads are generally solid, and are keyed on to the valve rod and secured home by stout nuts which are drilled and pegged in situ. Exact positioning is obtained by the introduction or removal of copper shims behind the heads when valve setting. Steam tightness is effected by the fitting of four or up to six narrow rings (the broad ring having generally gone out of favour for more or less obvious reasons) in accurately turned grooves around the circumference. (See Fig. 48b). Alternatively the valve heads can be hollow, with a central boss ribbed to the head, in which case they are connected by a hollow tube between the heads which allows of free circulation of steam through the valve itself. Once again they must be given "lap" and "lead" for purposes of correct steam distribution, and they may be either of the "inside admission" or "outside admission" type depending on whether boiler steam is admitted to the cylinder between the valve heads, or over the outer edges of the valves respectively.

Considerable advantage lies with the former method, in these days of long lap and long travel valves, whereby the exhaust ports are made straight and direct, in that the valve spindle bushes do not require a gland and packing in the valve chest covers, as they are only in contact with exhaust steam. Piston valves are usually of high quality cast iron, a reasonable analysis being C. 3·3·3%, Si. 1·2-3%, Mn. 0·8-0·9%, S. 0·1% max., P. 0·5% max., which analysis is completed by a low phosphoric pig iron; cylinders, pistons, and liners are often cast from the same melt, and piston and valve rings are also of the same metallic composition. In order to reduce the weight of piston valves, light spider heads of

[5]For detailed and illustrated descriptions see *Locomotive Valves and Valve Gears*, by C. S. Lake and A. Reidinger.

[6]An ingenious valve in which the heads only are affixed to the valve rod, the bodies floating in the valve liners, thus acting as an automatic by-pass. A Russian invention used on some engines on the P.L.M. Railway of France, and on an 0-6-4 tank engine on the Metropolitan Railway in 1929.

drop forged steel on a small diameter spindle are in use, whilst the use of aluminium alloys has been considered, it must be remembered that the co-efficient of expansion is nearly twice that of steel, and its inordinate ductility gives rise to burring over of the ring grooves due to wear, with consequent ill effects,[7] which can only be countered by the use of an extended valve rod.

Whilst inside admission is favoured in modern practice, a notable exception is the engines of the former Southern Railway of the "Merchant Navy" and "West Country" classes in which outside admission valves are used due to the design of the "Bulleid" Valve gear[8] in which the valve is actuated by a rocking arm placed in the exhaust passage. There being no extended valve rod, the valve chest covers are plain.

Of the size of piston valves, it is general to-day to make them at least one-half of the piston (cylinder) diameter. Table I is according to Selby (Proc. I., Loc.E.),[7] whilst Table II gives a random selection of valves representative of British locomotive design practice.

TABLE I.—PISTON VALVE AND PORT DIMENSIONS (Selby).

Cylinder Dia. Inches	Piston Valve Dia. Inches	Steam Ports in Liners		
		Effective Length Inches	Width Inches	Area Square Inches
12 - 14	7	15	1	15
14 - 16	8	17½	1¼	21·9
16 - 18	9	20	1½	30
18 - 20	10	22½	1¾	39·3
20 - 22	11	25	1⅞	46·9
22 - 24	12	27½	2	55·0

[7]For a comprehensive treatment of the subject see *Steam Locomotive Design Data and Formulae*, by E. A. Phillipson.

[8]Patented by the designer O. V. Bulleid, C.M.E., S.R., 1937-1947, in which the valve gear is ⅜ full size, chain driven, and enclosed in an oil-tight gear case. See *Pocket Book for Mech. Engs.*, by D. A. Low (p. 673).

TABLE II.—COMPARATIVE PISTON VALVE DIMENSIONS (Actual).

	L.N.E.R. A.1	L.N.E.R. A.4	L.M.S. "Duchess"	L.M.S. Class 5	S.R. M. Navy
Clyinder Diameter	19″	18½″	16½″	18½″	18″
Valve Diameter	10″	9″	9″	10″	11″
Lead	⅛″	⅛″	¼″	·2735″	⅛″
Steam Lap	1⅝″	1⅝″	1¾″	1½″	1⅝″
Exhaust Lap	nil*	nil*	− 1/16″†	− 1/16″†	nil*
Max. Valve Travel	6 11/16″	5¾″	7 1/32″	6½″	6¼″

*Exhaust edge of valve, and port edge line in line.
†Negative lap = exhaust clearance in central position.

Ports.

Should be as direct and straight as possible, and proportioned so that the velocity of steam (exhausting) lies between 4,000 and 6,000 feet per minute for piston valves, up to 9,000 feet per minute for slide valves, and 12,000 for poppet valves.

Let V = Mean velocity of steam feet per minute.
S = Average piston speed feet per minute.
D = Cylinder dia. in inches.
a = cross sectional area of port normal to steam flow.

then

$$a \text{ (sq. in.)} = \cdot 7854 \ \frac{D^2 S}{V} \qquad \text{(i)}$$

For valve liners let D = cylinder dia. in., and
A = cylinder area sq. in.

then

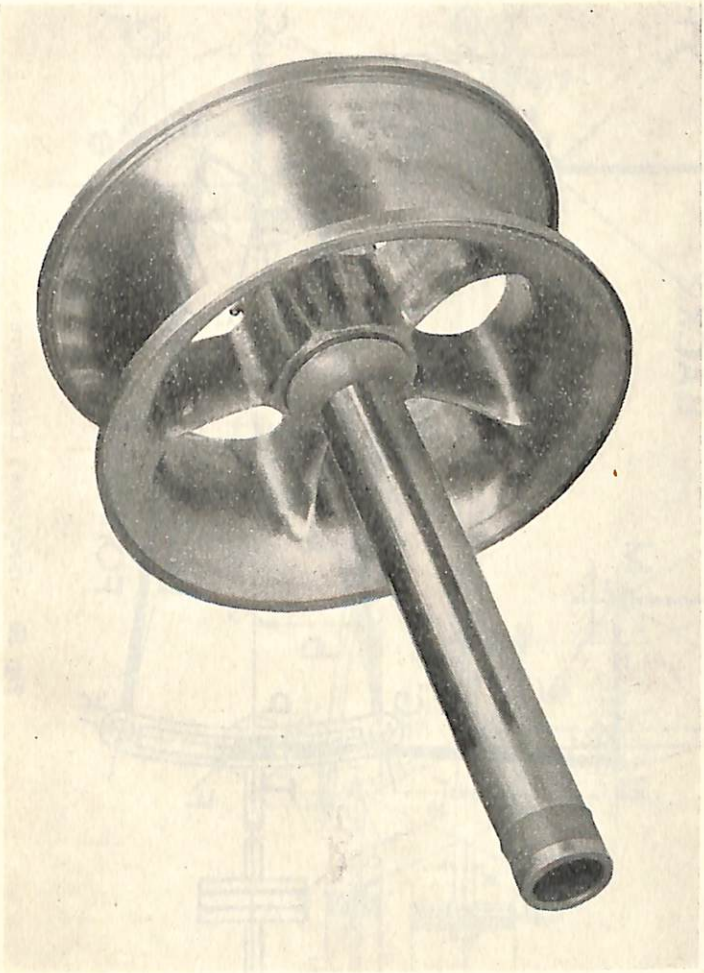
Area of ports in liner = 0·15A (Steam)
,, ,, ,, ,, ,, = 0·36A – 0·375A (exhaust)
Effective length of port = 16 × port width or 1·28D } (ii)

A less empirical method is based on volume swept by the piston, whence Port area = 1 sq. in. per 200 cu. in. of working cylinder volume. For slide valves :—

Steam port area = 0·095A
Exhaust port area = 0·19A – 0·21A
Length of Port (approx.) = 11 × Port width = 0·85D (British) } (iii)

or

1 sq. in. of Port area to 270 cu. in. of swept volume.



[*Courtesy A.L.E., Ltd.*

Fig. 49—British-Caprotti Valve Gear.
Five-Arm Poppet Valve with Spindle.
Scientifically designed for "streamlined" steam flow and maximum strength with minimum weight. Nitrided wear-resisting spindles maintain accuracy of fit and give long life.

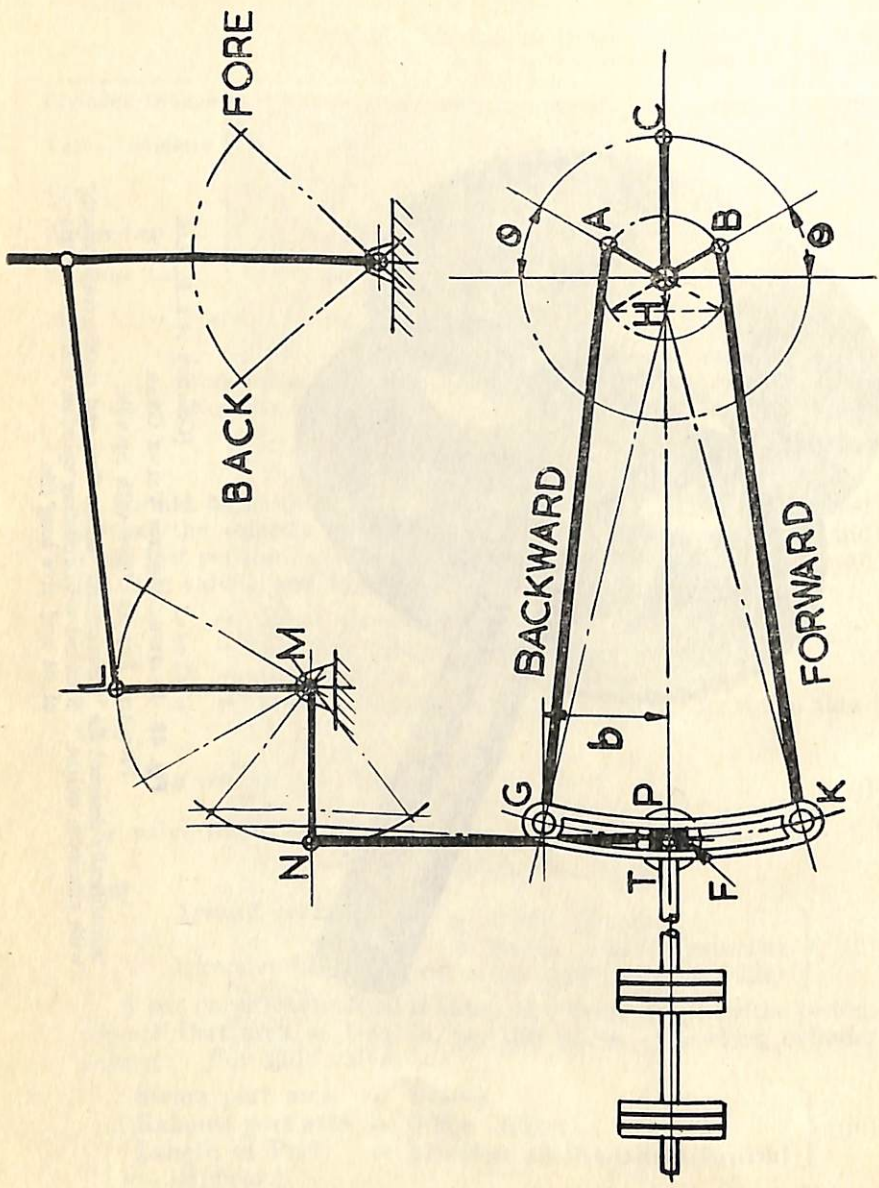
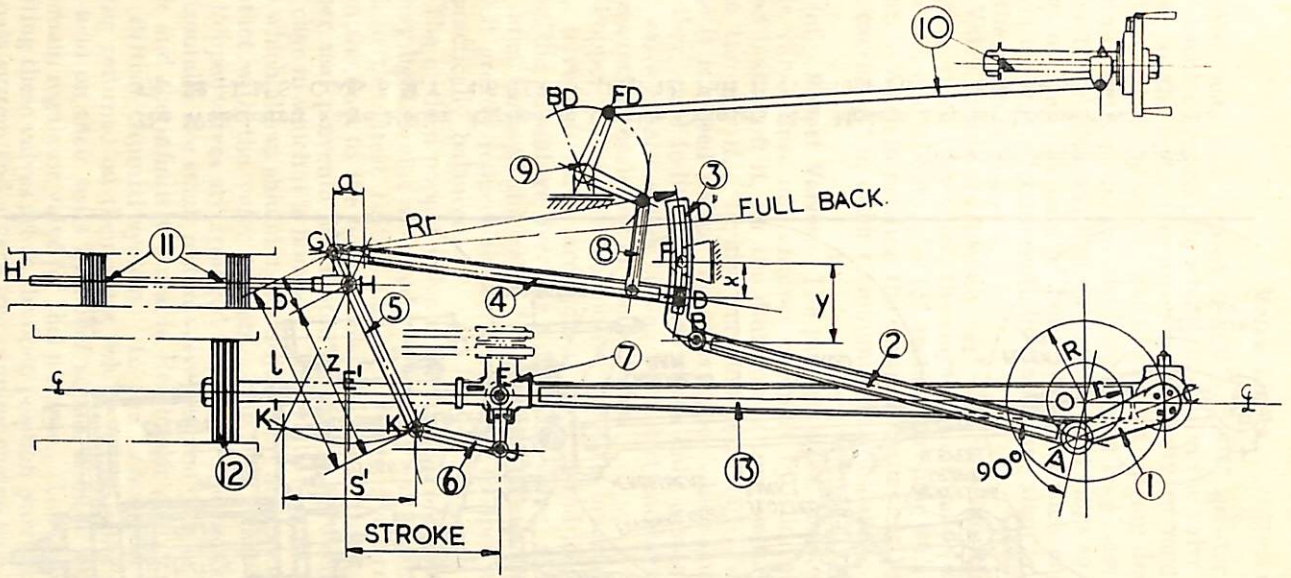


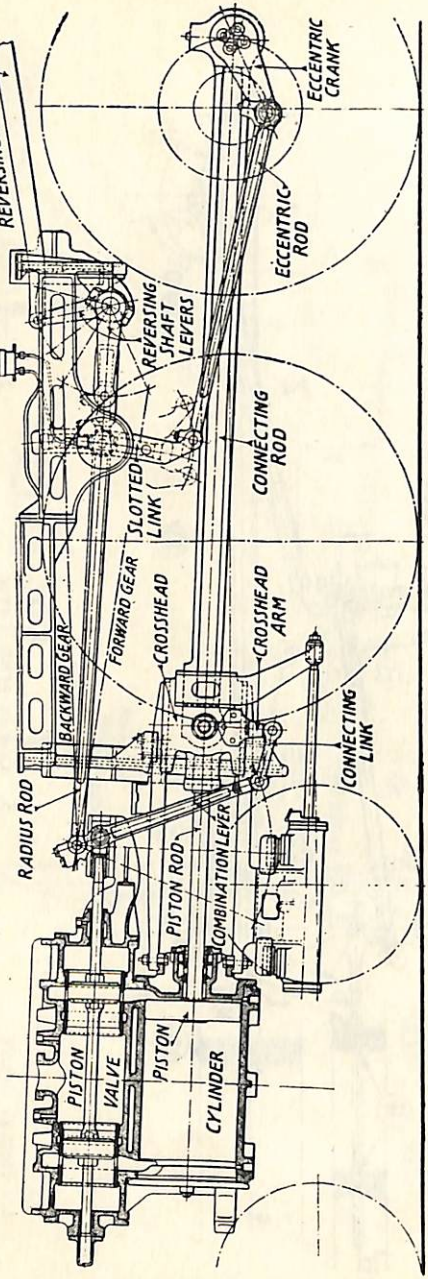
Fig. 50—Stephenson's Link Motion.



KEY.

1. Return Crank.
2. Eccentric Rod.
3. Expansion Link.
4. Radius Rod.
5. Combination Lever.
6. Union Link.
7. Cross Head.
8. Lifting Link.
9. Bell Crank and Cross Shaft.
10. Reversing Screw and Rod.
11. Piston Valve.
12. Piston.
13. Connecting Rod.

Fig. 51—WALSCHAERT'S VALVE GEAR.



[*Courtesy Railway Gazette.*

The Walscharets Valve Motion Applied to Outside Cylinders of a Modern Express Locomotive.

Fig. 52—L.M.S. Class 5 M.T., 4-6-0 Loco. (Fig. 12, Part 1) (Vacuum Pump, since Removed).

Valve Liners.

Are cast, and turned 0·003″ – 0·004″ greater diameter than the machined recess into which they fit in the valve chest, are bored taper at the ends, not exceeding 1 in 12 to facilitate entry and withdrawal of the valve, and to prevent ridging due to wear. The overall length of the parallel bore should be $\frac{1}{8}$″ less than maximum valve travel plus width of valve head over rings so that the outside rings overrun the parallel bore by $\frac{1}{16}$″ at each end in full travel. Ports are cut circumferentially and must align with the passages in the main casting.

Valve heads are usually $\frac{1}{64}$″ – $\frac{1}{32}$″ less in diameter than the liner in which they work, the rings which should be at least four in number, and about $\frac{9}{32}$″ × $\frac{11}{32}$″ section, keeping the assembly steam tight. When in position the circumferential gap in the ring should not exceed $\frac{1}{16}$″ in order to minimise steam leakage.

(iii) **Poppet Valves.**—The poppet valve, unlike the piston and slide valve, does not uncover the port by sliding over it, but is lifted bodily off it, for the duration of time required for steam to pass through it. It also remains stationary during the whole time it is closed unlike the other types which continue to move. Further, each individual valve serves only a single purpose, *i.e.*, one valve admits steam to one port, and another valve allows of its escape, at each end of the cylinder.

Poppet valves for locomotive use are invariably double-beat (*i.e.*, have two valve faces joined by a cylindrical portion) which gives the advantage over the single seat valve, of nearly doubling the area through the valve for a given lift, and more important still it greatly reduces the force required to open the valve which is practically balanced.

The modern poppet valve is machined from solid steel forgings, is very light, and is mounted individually on its spindle, is annular, and in the latest British Caprotti design five arms run radially from the boss to the body, whilst the flat bottom seat and conical upper seat prevents differential expansion between the valve and the cage in which it is housed. The valve spindle has a removable cap which allows shimming for adjustment and provides a hardened contact with the valve tappets. (Fig. 49).

Poppet valves are arranged in pairs at the ends of the cylinders, horizontally like small piston valves, in the case of the Lentz type gear and the infinitely variable Reidingear gear, and vertically in the British Caprotti system. In the Lentz gear the valves are spring returned on their seats, whilst in the other two gears they are held on their seats by boiler steam, a bleed being taken from the main regulator valve for this purpose. Thus when the engine is drifting these valves are in the open position providing an automatic by-pass system and a very free running engine as a result.

(3) Types of Valve Gears.

Are as about as legion as the valves they have been devised to drive. For descriptions of those gears other than the ones to be dealt with here the reader is referred to the extensive bibliography available on this subject ; for the purposes of this excursion into this diverting topic it will suffice to deal with (i) the Stephenson Link Motion ; (ii) Walschaerts Valve Gear ; (iii) Modern R.C. Poppet Valve Gears.

(i) **Stephenson's Link Motion.**—Reciprocating valve gears are either of the Link Motion or Radial Gear types, the Stephenson's gear belonging to the former group. It can be used with crossed eccentric rods, or open rods, inside admission, or outside admission, with or without a rocking shaft (which reverses the valve travel relative to the direction of motion of the gear). There being little virtue in needless complication, the simplest form will be considered, *i.e.*, having open rods, inside admission (for piston valves) without a rocking shaft.

From Fig. 50 it will be seen the gear consists of two eccentrics OA, OB mounted on the driving axle, and pinned to the extremities of a slotted link GK. This link is suspended by a lifting link TN from one arm of the bell crank LMN on the cross shaft, in such a way that it is free to swing about its mid-point P as it is driven by either eccentric. A die block F slides within the link GK and is pinned to the valve rod. The lifting link TN carries the trunnion bearing thus allowing the link freedom to swing. When the link GK is in its lowest position the valve rod is driven by the backward eccentric OA, when in its highest position, the valve rod is driven by the forward eccentric OB. It is shown in mid-position whence the valve will have minimum travel.

Theoretically the eccentric must precede or follow the crank by 90°. In actuality, in order to give lead to the valve, the eccentric is advanced by a small angle known as the angle of advance. Therefore, in Fig. 50, the forward eccentric follows the engine crank OC by an angle 90°—angle of advance, the backward eccentric therefore leads the crank by a similar amount. With this type of connection the lead of the valve is reduced as its travel is reduced, *i.e.*, by lifting or lowering the link GK toward the mid-position. With crossed eccentric rods the eccentrics must be at an angle of 90° + angle of advance from the crank ; in the rear dead centre position shown, and the lead in this case will increase as "notching up" to mid-position occurs.

The link GK should be of radius GH, and in mid-gear NM should be parallel to the longitudinal centre line of the gear ; further a line drawn vertically through P should bisect the versed sine of the arc in which N moves, in this position of the valve gear. NM

should be as long as possible, and not less than $\frac{AG}{4}$ and the length of link TN should be a maximum so that locus of P may be as nearly coincident as possible with the longitudinal centre line of the gear. In general all the links should be as long as possible, other than the throw of the eccentrics, and total length GK, to minimise the effects of angularity. In mid-gear, the valve is just open to lead, the approximate displacement of the valve from its mid-position is given by $r \sin \theta + r \left(\frac{b}{l} \cos \theta\right)$ where OA = OB = r, AG = BK = l.

(ii) **Walschaerts Valve Gear.**—An invention by a Belgian in 1844 whose name it carries, is probably the most extensively used gear in the world in the modified form in which it has become established. It is of the radial type and possesses a number of outstanding features, which are advantageous to it over almost every other gear, link motions in particular. It is :—

(*a*) Simple in conception and disposition.
(*b*) Equally suitable for inside or outside cylinders.
(*c*) If used externally an eccentric is dispensed with.
(*d*) Being a radial gear the lead remains constant for all positions from full-gear to mid-position.
(*e*) If well designed and well set, is an excellent steam distributor.[9]

From Fig. 51, it can be seen that the valve (which is mounted on valve rod HH[1]) derives its motion from two components, viz., a return crank AC having an effective throw of 2*r* and the cross head 7. The return crank 1 is set so that the motion it imparts to the valve is 90° ahead of or behind that received from the cross-head 7. The motion from the return crank 1 is transmitted by the eccentric rod 2 to the expansion link 3 which is mounted in trunnion bearings F rigidly fixed to the engine, about which it is free to swing. A die block D transmits this motion to the radius rod 4 and ultimately to the combination lever (or pendulum link) 5. Reversing and alteration of cut off is effected by raising or lowering the die in the link by means of suitable lifting gear 8 and 9 under the control of the driver. When D is as shown the gear is generally fully forward, when at F it is in mid-gear, and when at D[1] the gear is then fully reversed.

It will be seen that the eccentric's contribution to the motion of the valve varies directly as the distance D is from F from a maximum at D or D[1] to zero at F.

[9]The World's speed record for steam of 126 m.p.h., held by L.N.E.R. A4, 4468 "Mallard," which is fitted with external Walschaerts Valve Gear.

The effective radius of the link 3 (Rr) is equal to the effective length of the radius rod 4, *i.e.*, GD. When the cross head 7 is at the end of its stroke, *i.e.*, crank on either dead centre, the link is at the mid-point of its swing, due to the fact that the motion derived from the return crank 1 is 90° out of phase with that derived from the cross head. The link should not have a total swing of over 45° (*i.e.*, 22½° either side of its vertical position) otherwise the die block tends to ride up the link due to the force required to reverse the piston valve, causing a kick which may be harmful.

Thus the lead is entirely dependent on the stroke of the cross head, and as this is constant the lead itself is also constant for all cut offs, *i.e.*, all positions of the Die Block D in the link 3. This is a feature of all radial gears. The combination lever 5 is so proportioned that the valve travel "a" derived from it equals 2 (lap + lead), and the ratio

$$\frac{b}{l} = \frac{2\ (\text{lap} + \text{lead})}{\text{Piston Stroke}} \qquad \text{(iv)}$$

The configuration shown is for an inside admission valve, for outside admission, the points G and H are reversed but the ratio b/l still obtains.

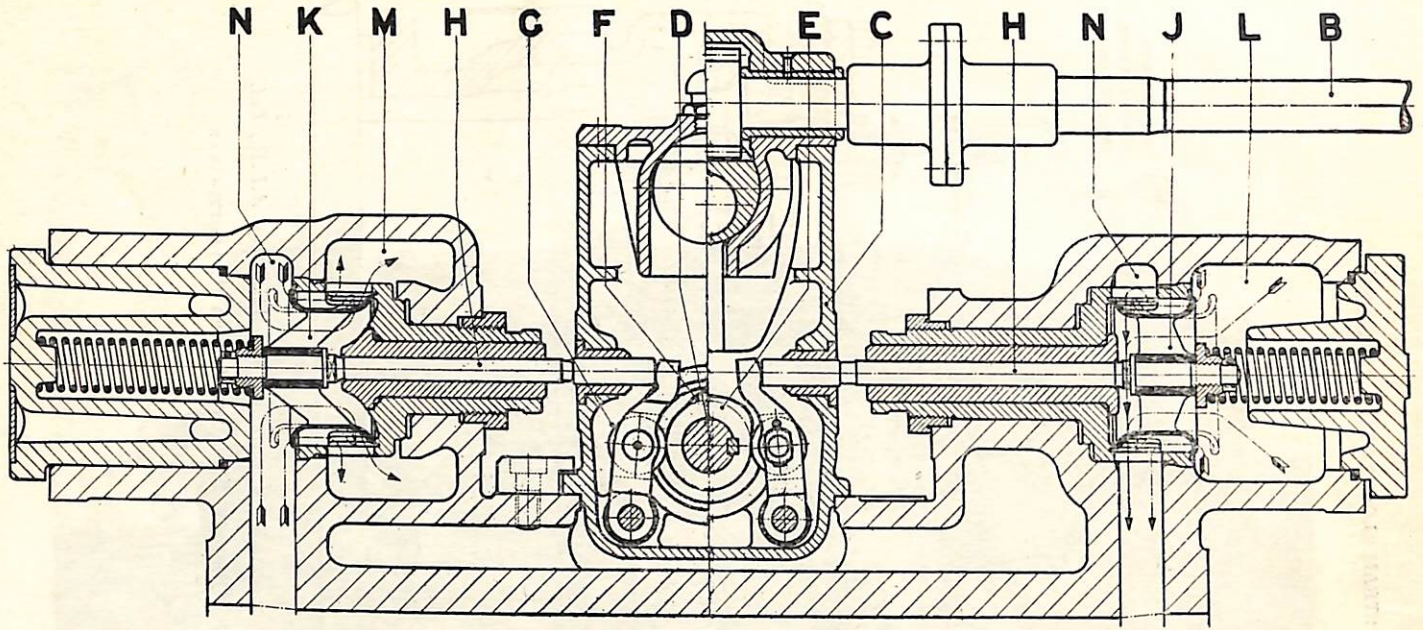
The return crank pin A is set at 90° to a line drawn through the link foot pin B and the centre of the driving axle, point O, when the crank OC is on dead centre and the link at mid-swing, hence the return crank need not actually be 90° from the main crank, although as aforementioned the motion derived by the valve from it is always 90° out of phase with that derived from the cross head. In forward gear it is general to arrange the radius rod to follow the eccentric rod as shown, thus in reverse gear the radius rod is 180° out of phase with the eccentric rod, thus reversing the engine.

Among points to remember are :—

(i) The length of AB should be a maximum to reduce the effects of angularity (to which end a link foot is provided) and should be at least seven, and up to 10 or 12 times the diameter (2r) of the return crank circle.

(ii) It is vastly preferable (if exigencies of design allow) to raise and lower the radius rod by a slotted backward extension in which a bronze lifting block pinned to the forked arm of the bell crank can ride (Fig. 52). Failing this the pin joint of 8 to 4, should be as near the link 3 as possible, and the length of 8 should be as long as possible to reduce slip of the die block D in the link 3 caused by the pin joint of 8 swinging in an arc, and not moving in a straight line causing the die block end of the radius rod to rise and fall a distance equal to the versed sine of the arc of swing of radius equal to 8.

(iii) The combination lever 5 must be perpendicular to the piston stroke at mid-stroke, and its length such that the point G

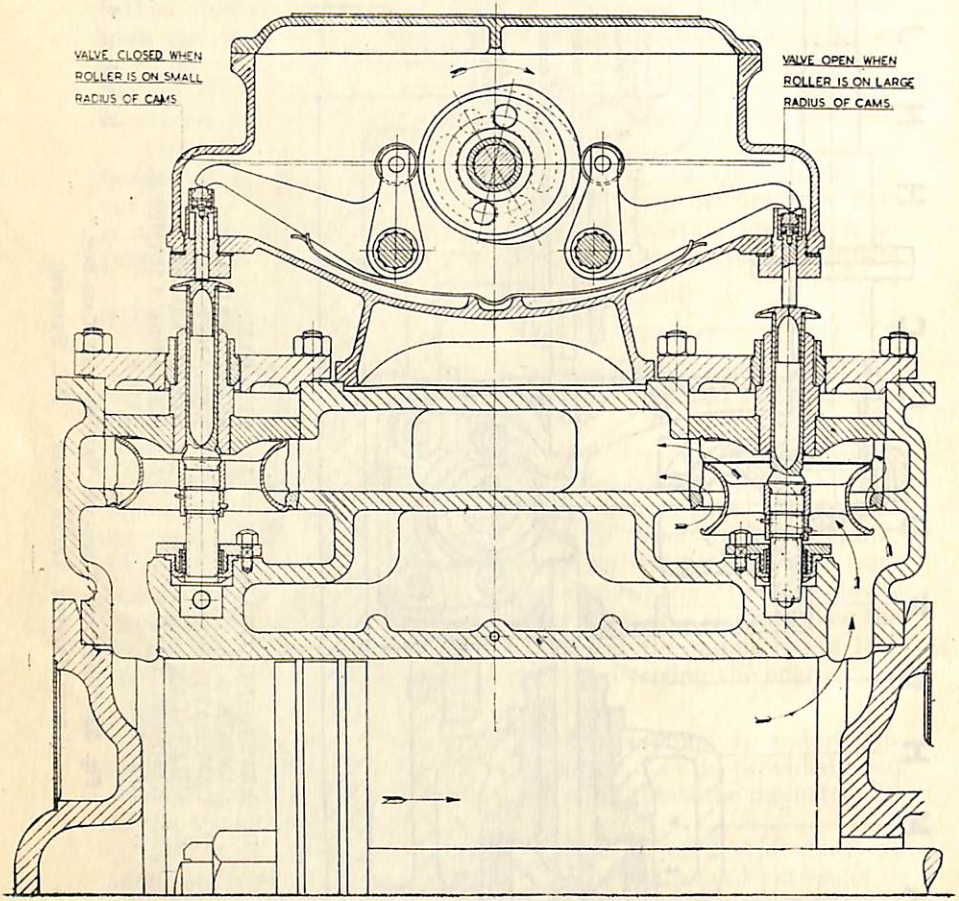


KEY.

B. REVERSING GEAR.
C. CAMBOX.
D. CAMSHAFT.
E. INLET CAM.
F. EXHAUST CAM.
G. ROLLER.
H. SPINDLE.
J. INLET VALVE.
K. EXHAUST VALVE.
L. STEAM CHEST.
M. EXHAUST CHAMBER.
N. PORT.

[*Courtesy A.L.E., Ltd.*

Fig. 53—"Section Through Valves and Cam-Box."
(Lentz Poppet Valve Gear. R.C. System).



[*Courtesy A.L.E., Ltd.*

Fig. 54—Section Through Exhaust Valves (Caprotti-Gear).

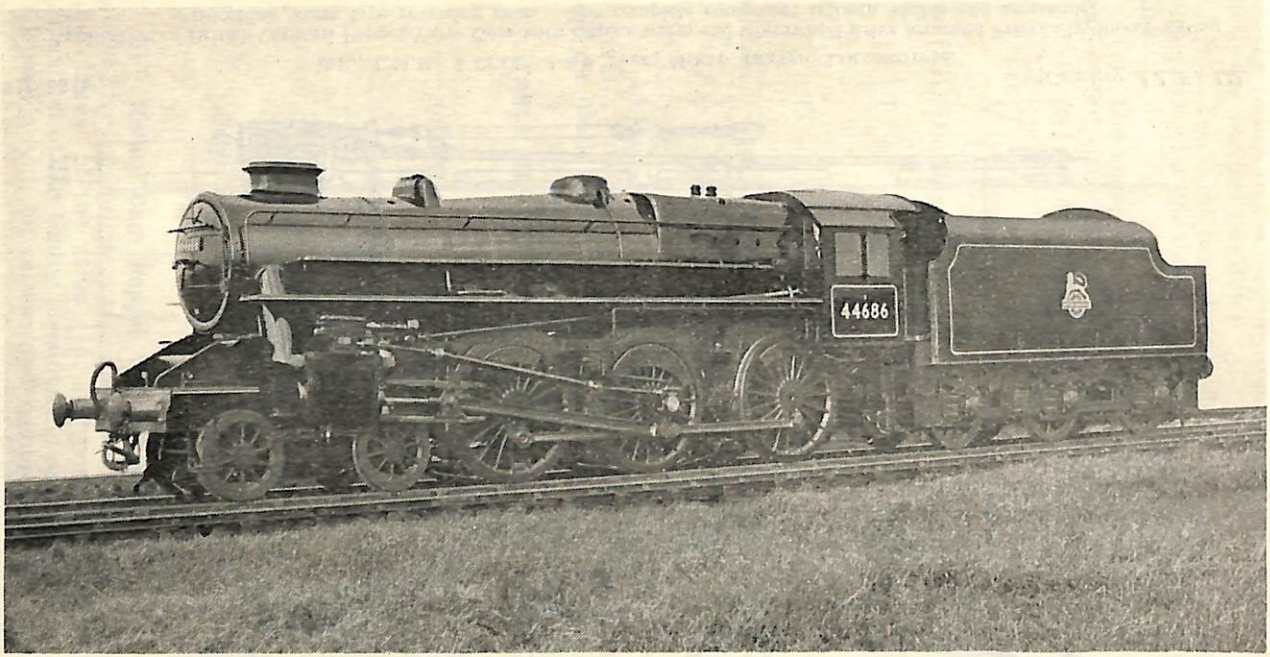


Fig 55 (a). *[Courtesy British Railways and A.L.E. Ltd.*

B.R., L.M.R., Class 5 M.T. Locomotive with British-Caprotti Valve Gear, Double Chimney and (S.C.) Self Cleaning Smokebox. (For dimensions see Fig. 12). Photograph shows the ultimate development of this outstanding design, comprising in all eleven variations and 842 locomotives built between 1934 and 1951 for both L.M.S.R. and B.R. to the designs of W. A. Stanier and H. G. Ivatt (C.M.E., L.M.S. & L.M.R., 1932-44 & 1945-1951).

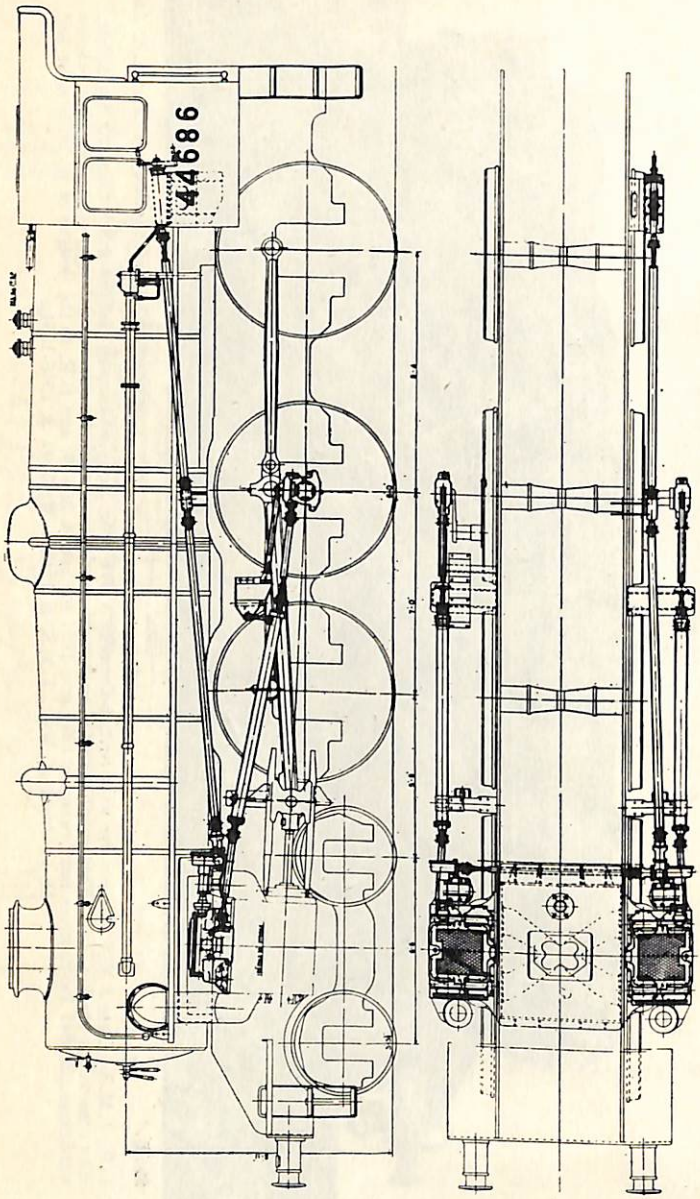


Fig. 55 (b). B.R., L.M.R., 5 Class, 4-6-0 Type, Mixed Traffic Locomotives. [*Courtesy A.L.E., Ltd.*

Application of British-Caprotti Poppet Valve Gear with duplex worm and wormwheel drive arranged externally, incorporating self-locking worm type reversing gear. The complete equipment is both visible and accessible.

on the radius rod is normal to a straight line KK^1 when the link is in the vertical position.

(iv) The cross head link 6 should swing an equal amount on either side of a line drawn through J parallel to the cylinder centre line.

(v) The angle of swing of the combination lever 5 should not exceed 30° either side of the vertical position.

(vi) Travel of valve in full gear (symbols as used in Fig. 51).

$$\sqrt{\left\{\left(\frac{b}{b+z}\right)\times S\right\}^2+\left\{\left(\frac{x}{y}\right)\times 2r\right\}^2} \qquad (v)$$

(vii) Die Block travel in full gear is given by

$$T = \frac{S \times \sqrt{V^2-a^2}}{S+a} \text{ for inside admission} \qquad (vi)$$

and

$$T = \frac{S \times \sqrt{V^2-a^2}}{S-a} \text{ for outside admission} \qquad (vii)$$

where

T = travel of die block horizontally.
V = travel of valve horizontally.
S = piston stroke.
a = 2 (lap + lead).

(viii) Finally, Table III gives relative positions of return crank and die block for the usual variations of Walschaerts Gear.

TABLE III.—DETAILS OF WALSCHAERTS VALVE GEAR.

Admission	Position of Block in Link in Full Fore Gear	Position of Return Crank or Eccentric Relative to Main Crank
Inside	At bottom	Follows main crank
Inside	At top	Leads main crank
Outside	At bottom	Leads main crank
Outside	At top	Follows main crank

(iii) **Rotary Cam Poppet Valve Gears.**—The ideal valve gear would give (*a*) infinite variability of cut off; (*b*) rapid openings and closings of the valve; (*c*) full port opening at all cut offs; (*d*) control of the timing of admission and release to a nicety, and independently; (*e*) release and compression independent of cut offs. Radial and link gears cannot fulfil all the above conditions,

but Poppet Valves operated by the modern rotary cam gears come very near to it.

(iv) **The Lentz R.C. Poppet Valve Gear.** (Fig. 53).—Consists of four valves placed horizontally above the cylinder extremities and operated by a camshaft driven at driving wheel speed (but revolving in the opposite direction), housed in a cambox which is bolted above the cylinder between the valve pockets. The drive is taken from a return crank through external gear boxes suitably suspended from the engine frame, and thence by external shafting with universal joints to allow for relative movement between the engine and driving wheels, to a worm reduction gear mounted on the end of the camshaft.

One group of cams is required to operate the steam valves and another adjacent group the exhaust valves, for forward and backward running. The cams are arranged between horizontally opposed valves, so that each cam operates two valves during one revolution. A full forward position is arranged at one end of each group of steam and exhaust cams and reversal is effected by sliding the cams and camshaft transversally across the engine sufficiently for the cams at the opposite end of the cam shaft to take up the driving position. At intermediate points between these extreme positions, there are additional cams giving variation of events for the steam and exhaust valves.

Between each set of forward and reverse cams are Bye-Pass Cams, which are circular rings of sufficient diameter to open and hold open, both steam and exhaust valves, thus providing a free running engine when drifting.

Between the cams and the valve tappets are situated the intermediate levers, one end of which is pivoted on a fulcrum, the free end being in contact with the valve tappet or the valve spindle itself, whilst between the two ends is pinned a roller which makes contact with the cam. The cam rotation causes the roller to move outwards, and by lever action this movement is increased in speed and magnitude at the valve. One lever is required to operate each valve.

Transverse movement of the camshaft is effected by rack and pinion, the former carrying a stirrup which engages the cam shaft, while the latter through suitably jointed reduction gearing is under the control of driver in such a way that the rotation of a handle or hand-wheel enables various cut off workings or complete reversal to be selected. It should be obvious that there must be a cambox for each cylinder, and that their reversing mechanisms must be connected so that identical events occur in each cylinder. It is customary to provide two sets of external driving gear. The valves in the Lentz gear are maintained on their seats by adjustable

springs housed in the valve covers. This type of gear is now known as the "A.L.E." - "R.C." Poppet Valve Gear.[10]

(v) **The British Caprotti Gear.**—In the modern steam locomotive the development of tractive effort at all speeds, is not so much a question of steam supply limited by the output of the boiler, as of the efficient use of the steam in the cylinder, the criterion of which is the mean effective pressure. To maintain this the steam should flow unobstructed through ports and passages of generous proportions, and further to make the most of the expansive properties of the steam, correct relationship and accurate timing of valve events are imperative.

In the British-Caprotti Valve Gear the four poppet valves and their individual cages are accommodated at the ends of the cylinder, but in this instance vertically. An individual cambox is bolted above the cylinder casting containing the camshaft with its spiral splines and scroll assemblies, cams, reversing crankshaft and reversing rods, and intermediate rocking levers which actuate tappets placed in a vertical plane. The tappet ends protrude through the tappet cuffs which are now cast integrally with the cam box and are in contact with the ends or caps of the valve spindles. (Fig. 54).

The exhaust intermediate lever is in the form of a bell crank carrying a single roller at one end, the other end actuating the tappet. The steam intermediate lever is similarly shaped but carries a swing beam arrangement with a cam roller at each of its ends, one of which rides on the outside cam the other on the inside cam, which control the cut off and lead respectively.

To vary cut off independently of lead it is necessary to move the outside scroll along the camshaft whilst holding the inside scroll. As the scrolls are moved along the camshaft the spline configuration causes them to move with relative angular displacement thus altering the valve timing. The moving of the scrolls to the other extremity of the camshaft causes reversal of the engine.

The drive for the gear can be taken off externally by return cranks, or internally by a two-piece crown wheel (bolted to the axle) and bevel, in a suitable housing driving through jointed shafts to a cross driving gear to which the camshafts are coupled, it can be further arranged to drive two, three, or four cylinder engines with cylinders in line or offset, the camshaft revolving at driving wheel speed but in the reverse direction, on one side of the locomotive and in the same direction on the other. Thus the valve gear is not "handed."

The reversing gear is of the irreversible worm type, which has been introduced to absorb any reactions arising from the poppet

[10]Manufactured by Associated Locomotive Equipment Ltd., makers of the British-Caprotti Gear.

valves, ensuring both precision accuracy, and smoothness, in operation. The reversing hand-wheel and screw can be supplied in accordance with any standard practice, so that from the driver's point of view there is no apparent difference in the physical operation of poppet valve geared locomotives. (Fig. 55*b*).

The valves are maintained on their seats by boiler steam taken from the regulator, thus when coasting with closed regulator, all valves fall away from their seats, thereby ensuring a full area by-pass arrangement, which provides a free running engine, and minimises carbonisation of valves and cylinders.

In conclusion, the following table[11] gives some idea of gain in M.E.P. obtained with use of Caprotti gear. It should also be mentioned that as well as considerable use in this country primarily by the former L.N.E.R., the L.M.S., and now by British Railways, Rotary Cam Poppet valve gears have been used on Indian Railways, South African, Belgian, Argentine, Alsace Lorraine, Coras Iompair Eireann, Egyptian Republic, Malayan Railways and others. Recent examples in this country are 20 engines of the ubiquitous Class 5 Mixed Traffic Locomotives of the former L.M.S. (Fig. 55) and the recent Standard Class 8 British Railways Locomotive No. 71000 *Duke of Gloucester*. (See Frontispiece, Part I).

TABLE IV.

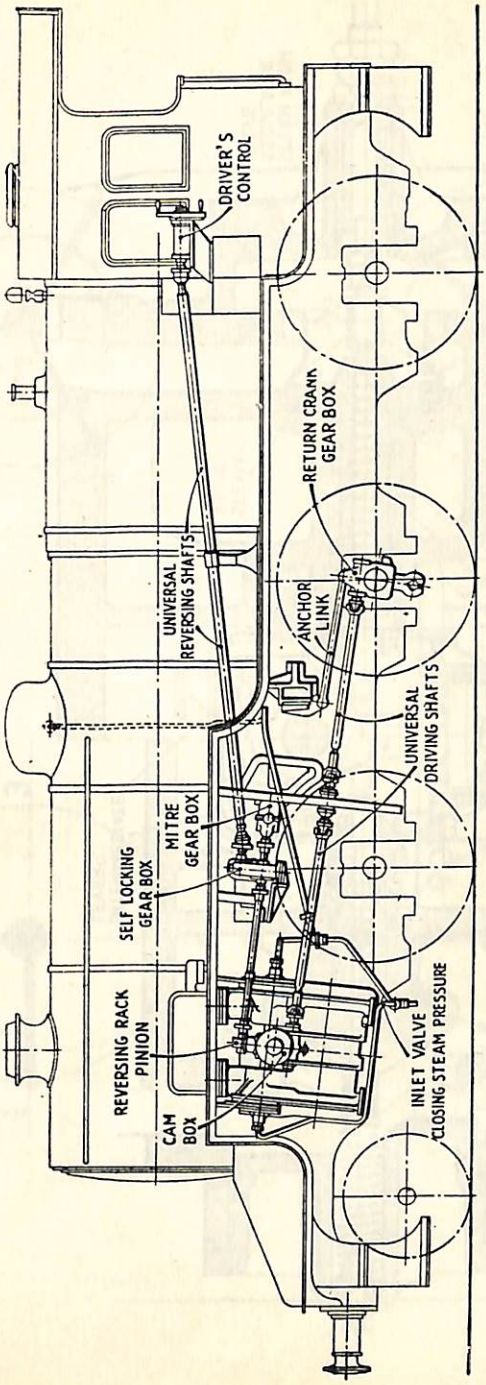
INCREASE IN M.E.P. WITH THE USE OF BRITISH-CAPROTTI GEAR.

Type of Gear	Boiler Pressure	Speed	M.E.P.
Stephenson	225 lb./sq. in.	53 m.p.h.	47·6 lb./sq. in.
Walschaerts	220 ,,	47 ,,	58·9 ,,
British-Caprotti	218 ,,	48 ,,	74·2 ,,

(vi) **The Reidinger R.C. Valve Gear.** (Fig. 56).—This bears the name of its designer[12] and greatly resembles the "Lentz" type gear, but with certain marked differences. The situation and arrangement of valves, driving gear and cam boxes is similar, but the intermediate levers actuating the steam valves in this case carry two rollers on a small beam, as in Caprotti practice, one running on the "pre-admission" cam, and the other on the "cut off" cam. The steam valves are retained on their seats by boiler steam from the regulator valve actuating on the outer ends of their spindles, whilst the exhaust valves are maintained on their seats by springs.

[11]By kind permission of A.L.E. Ltd., makers of the British-Caprotti Gear.

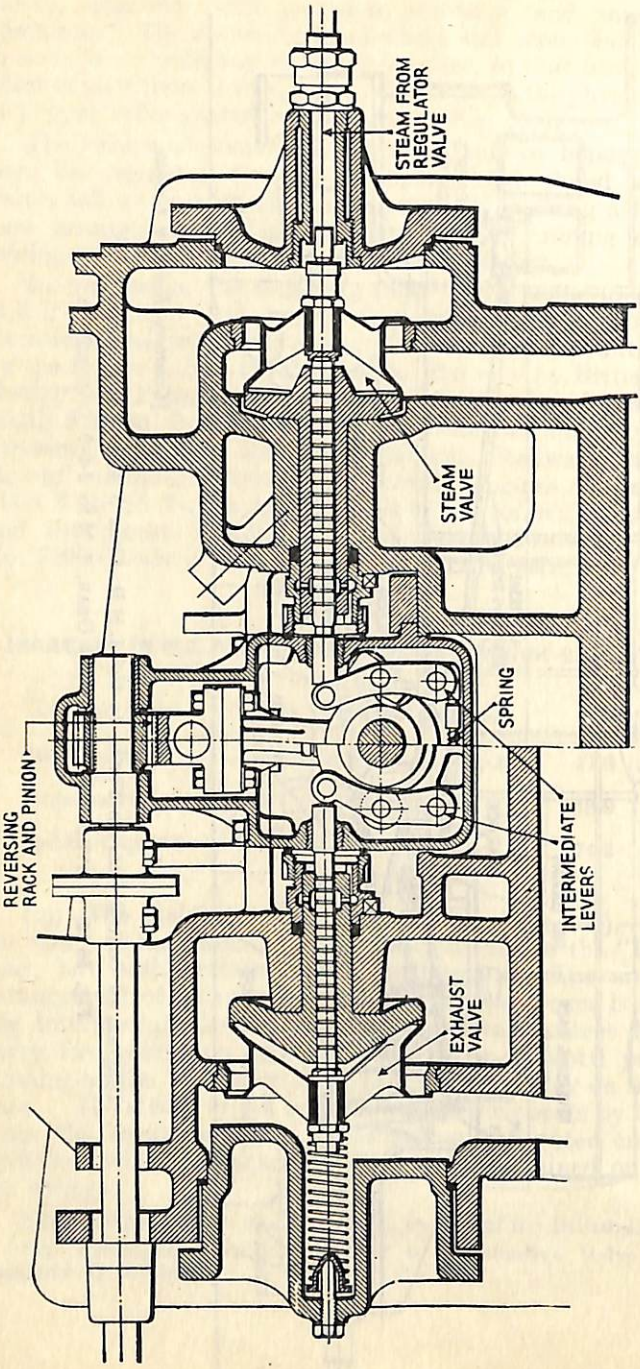
[12]A. Reidinger, Technical Adviser to Locomotive Valve Gears Ltd., suppliers of the gear.



[*Courtesy Railway Gazette.*

Diagram of Control Gear as Fitted to the L.M.R. Class 5 M.T. Locomotive.

Fig. 56 (a)—Layout of Reidinger Gear, L.M.S. 2-6-0 Locomotive.



[*Courtesy Railway Gazette.*

Arrangement of Steam and Exhaust Poppet Valves.

Fig. 56 (b)—Section Through Valve Box. Reidinger Gear.

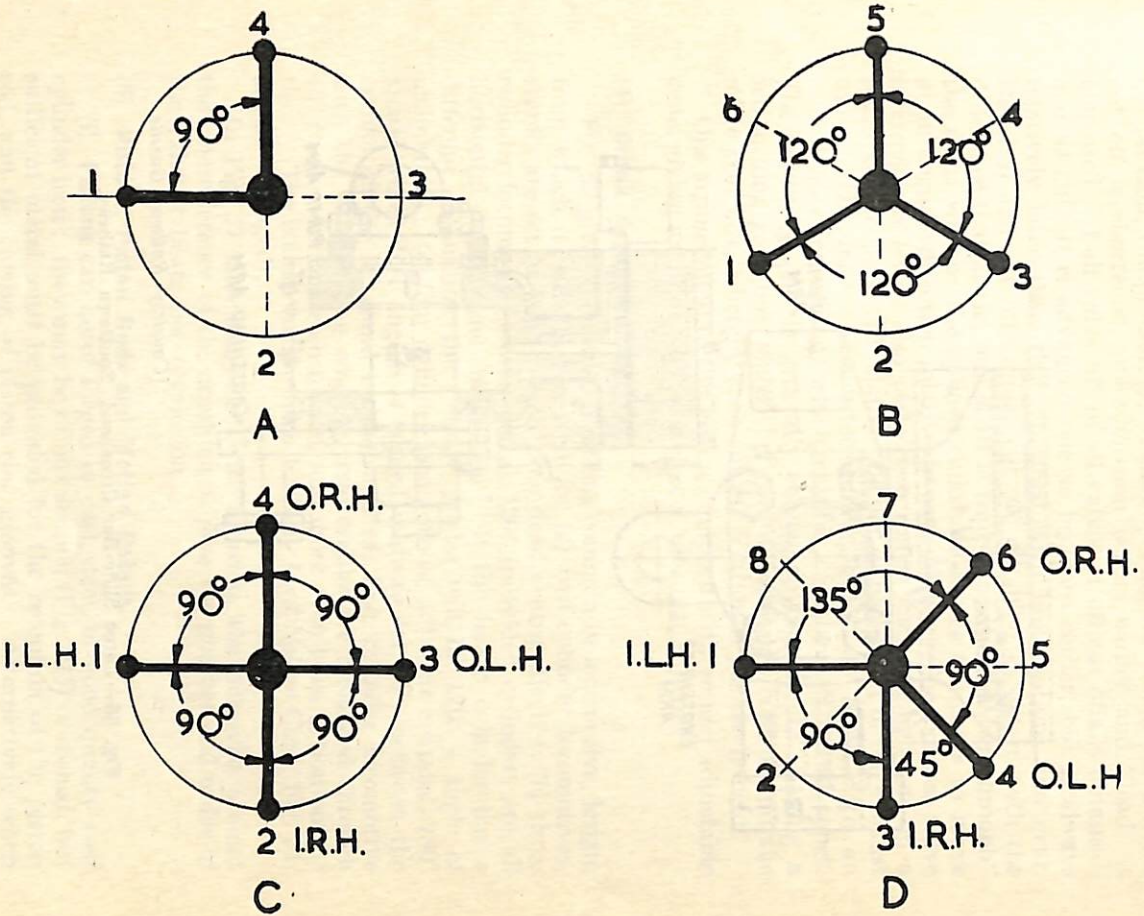
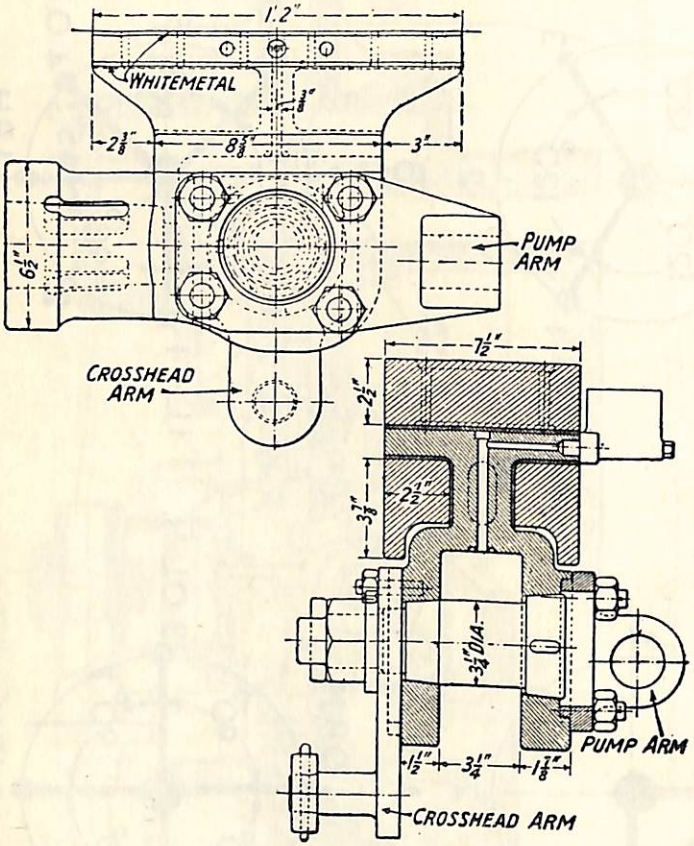


Fig. 57—Crank Arrangements.



[*Courtesy Railway Gazette.*

Fig. 58—Three Slide-Bar Crosshead, Southern Railway.

The exhaust valves are at all times in contact with and actuated by the exhaust cam, whilst all steam valve events are based on the cut off cam, the pre-admission cam opening the steam valves. As cut off is shortened, pre-admission occurs earlier and "lead" is increased. Full valve lift of all valves at all cut offs is obtained, and as cut off is advanced release becomes earlier, but is always relatively later than that obtained with link motions and gear. Reversal and cut off adjustment are obtained as before with the difference that as the cam shaft is translated it is "drawn through" the cams whence the splined camshaft causes the cams to move in an angular direction relative to each other. The exhaust valve intermediate levers also carry two rollers each, though not on a beam, due to the small relative angular displacement of the exhaust cams. The inlet valve intermediate levers are fitted with an inter-connecting spring, which ensures cam contact at all times. The reversing gear is generally similar to the "Lentz" having a self-locking gear box placed between the driver's control and the mitre gear box, whereby each cut off position is positively held.

One arrangement of this gear is equipped with vernier adjustable cams giving an infinitely variable cut off arrangement.[13]

(4) **Crank Arrangements.**

Fig. 57 gives the crank positions common in locomotive design, from which it will be seen that for (*a*) two cylinder locomotives, cranks are set at 90° (to overcome dead centre effect) ; (*b*) three cylinder locomotive cranks set at 120° (provided cylinders are all horizontal or at same inclination ; if the inside cylinder has a different inclination the relative angles will be 120° ± angle of inclination) ; (*c*) four cylinder locomotives, adjacent cylinders 180° apart, whilst RH engine is placed midway (at 90°) between the cranks of the left-hand engine, and (*d*) four cylinder locomotive with the 135° setting whereby two cranks are advanced through 45° to give a more even torque, and more even blast action on the fire. (The cranks on Southern Railway *Lord Nelson* Class, Fig. 31, are so arranged).

In Fig. 57, assuming clockwise rotation, the numerals around the circumference of the crank circle give the number and order of the exhaust beats per revolution.

(5) **Pistons, Piston Rods and Motion Details.**

(i) **Pistons** can be of forged or cast steel, or good quality cast cylinder iron ; they may be of box section (Fig. 21*b*) or dished, but sufficient metal must be provided for the reception of the piston rod, and the cutting of piston ring grooves. Alternatively when

[13]As fitted to one of five L.M.S. 2-cyl. class 5, 2-6-0 Horwich built 1928 locomotives which are fitted with this gear. See *Railway Gazette*, 20/3/1953, or *The Locomotive*, April, 1953.

using steel, the piston head and rod can be forged in one piece. The boss must be bored (usually taper) to receive the piston rod, the head usually being finally tightened home against a shoulder by a large nut, drilled and pegged in situ.

For dished pistons thickness of metal is given by Unwin as :—

$$t_b = (D + 15)(0{\cdot}0012\sqrt{p} + 0{\cdot}012) \quad \text{(viii)}$$

and $$t_r = (0{\cdot}68 - 0{\cdot}002D)\, t_b \quad \text{(ix)}$$

where D = piston diameter in inches.

p = greatest pressure difference on sides of the head in lb./sq. inch.

t_b = thickness of head near the boss in inches.

t_r = thickness of head near the rim in inches.

The diameter of boss = 1·9 × diameter of piston rod in the boss. No uniformity exists as to barrel clearance, the difference in diameter of piston and barrel varying from $\frac{1}{64}''$ up to $\frac{1}{8}''$. When piston and rod are integrally forged, the piston should be provided with a detachable rim, as the scrapping size of the head is always attained earlier than that of the rod. This should be of a ferrous metal and secured by rivets, screws, or bolts.

Two or preferably three narrow rings are fitted for steam tightness (the slit in which should be made diagonally) with a radial depth of about $\frac{9}{16}''$ or $\frac{5}{8}''$ or from 0·025 × Dia. to 0·038 × Dia. The width should be about $\frac{3}{8}''$, and the end rings so located that with total allowance for big end brass wear taken up, they over-run the ports by no more than half their width. The rings are usually of the same composition as the cylinder barrel (*i.e.*, cylinder iron), this being most suitable for engines using superheated steam.

(ii) **Piston Rods** are usually made from "class D", B.S.S. Report No. 24, Spec. 8, steel having an U.T.S. of 40-45 tons per square inch, 20-25% elongation, and Yield Point of not less than 20 tons per square inch. The taper at the piston head may vary successively from 1 in 3 to 1 in 8 (usual fig. 1 in 6) and for the cottered joint at the cross head 1 in 12, 16, 20 and 1 in 24.

They are of necessity greater in diameter than the requirements of loading in order to allow for re-machining to remove the ovality obtained due to wear, the stress in the rod due to steam loading of the piston being usually less than 3 tons per sq. in. of cross sectional area. The stress induced due to the taper and screw attachment at the piston, and the taper and cotter attachment at the cross head is usually taken as 5-6 tons per square inch.

In determining the required diameter of the reciprocating portion of the rod it can be considered as being a strut (Case IV) rigidly fixed at one end (cross head) and guided laterally at the other, whence by the application of Euler's Theorem the effective length "L" is 0·7 × length of a case I strut, but having regard to the

floating nature of the support afforded by the gland, piston ring clearance, etc., it is more reasonable to consider it as a case I strut with jointed ends. Then adopting the Rankine Formula we have :

$$\frac{\pi}{4} D^2 p = \frac{W}{S} = \frac{fc A}{1 + \left(\frac{fc}{\pi^2 E} \times \frac{L^2}{K^2}\right)} \quad (x)$$

where D = cylinder diameter in inches.
p = greatest pressure difference on either side of the piston lb./sq. in. (this to include over compression and pressure at which relief valves operate).
W = crushing load on the rod in lb.
S = factor of safety.
fc = ultimate compressive strength of rod in lb./sq. in.
A = cross sectional area of the rod in sq. in.
E = Young's Modulus for the material in lb./sq. in.
L = overall length of rod in inches.
K = radius of gyration of the rod in inches.

The value of fc should be based on the Elastic limit in view of the alternating loads, and should not exceed one-third of it. The value of K^2 for a solid rod is $\frac{d^2}{16}$ (d = dia. of rod in.) and for a hollow rod $\frac{d^2_o - d^2_i}{16}$ where d_o = outside dia. in inches.

d_i = inside dia. in inches.

An empirical rule is to make $d = 0{\cdot}165\ D$ (xi

but this rule, it will be observed, does not take into account the slenderness ratio $\frac{L}{K}$ of the rod, and hence it is not being considered as a strut.

Continuing from (x) where $\frac{\pi}{4} D^2 p = \frac{fc A}{1 + \left(\frac{fc}{\pi^2 E} \times \frac{L^2}{K^2}\right)}$

we can say $p_o = \frac{W}{SA} = \frac{fc}{1 + a\left(\frac{L}{K}\right)^2}$ (xii)

where p_o = mean intensity of stress in the rod.
W = crushing load on the rod.
S = factor of safety.

and for values of fc from 3 tons/sq. in. to 36 tons/sq. in., the value of "a" will vary from 1/750 to 1/9,000, but it will be necessary to know the value of "a" corresponding to the value taken for fc which should not exceed $\frac{1}{3}$ the value of the elastic limit.

The threaded portion of the rod is subjected to pure tension, the value of whose stress based on the area of root diameter of the thread should give a factor of safety of 5 - 8. The crest diameter of the thread should be $\frac{3}{8}$" - $\frac{5}{8}$" less than the body diameter of the rod (for rods of 3 in. dia. and over) but considerably less, *i.e.*, about $\frac{1}{8}$ in.-$\frac{1}{4}$ in. less diameter with rods about 2 in. dia. body. About six threads per inch should be employed.[14]

Working stresses usually employed in piston rod design are :—

(i) Material surrounding the cross head cotter 5 T.P.S.I.
(ii) Area of cotter resisting shear 4 T.P.S.I.
(iii) Bearing pressure of cotter 8 T.P.S.I.

For a full treatment of cottered joints[15] two references are given.

(iii) **The Cross Head** is made to receive the piston rod end by means of a cottered joint, and the little end gudgeon pin, whose centre in end elevation, in position, should be co-planar with that of the piston rod. It may have two or more sliding faces, usually bronze slippered and white metal covered, it can be of the single bar type, two bar type, or three or four bar type, all but the two bar type usually being underhung. The gudgeon pin is introduced from the frame side of the cross head (on outside cylinders) and tightened home by means of a large nut prevented from turning by a split cotter. (Fig. 58).

Invariably made from steel it can be forged or cast, and subsequently machined. The size of the gudgeon pin is determined rather to give a reasonable little end bearing pressure than by consideration of strength, and this should be from 4,000 - 6,000 lb./sq. in. of projected area, *i.e.*, $l \times d$.

Where l = length of pin between supports.
Where d = pin diameter.

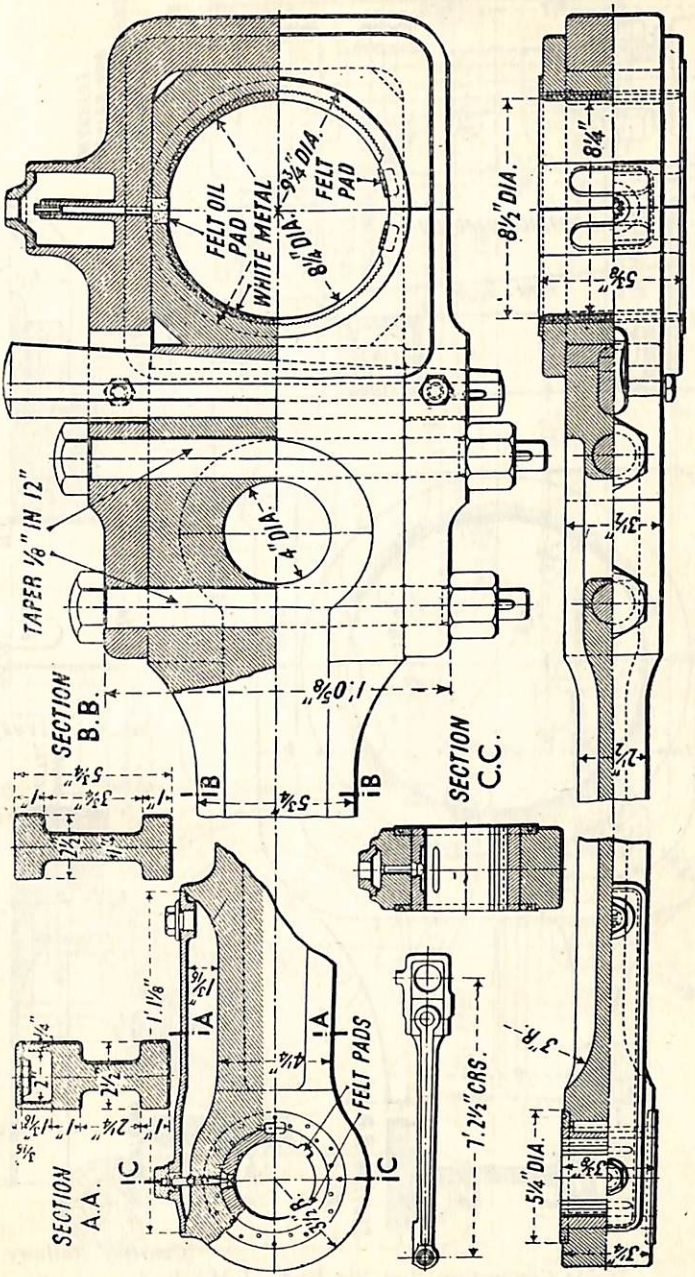
Having determined l and d, the strength of pin can be checked

$$\text{by M max.} = \frac{Pl}{8} = \cdot 098 d^3 f \qquad \text{(xiii)}$$

where P = max. piston load (lb. or tons).
f = max. working stress due to bending (lb. or tons to suit P) a representative value of f is 6,500 lb./sq. in.

[14]For table of Piston Rod dimensions see *Loco. Engineers' Pocket Book*, p. 131, 42nd edition.

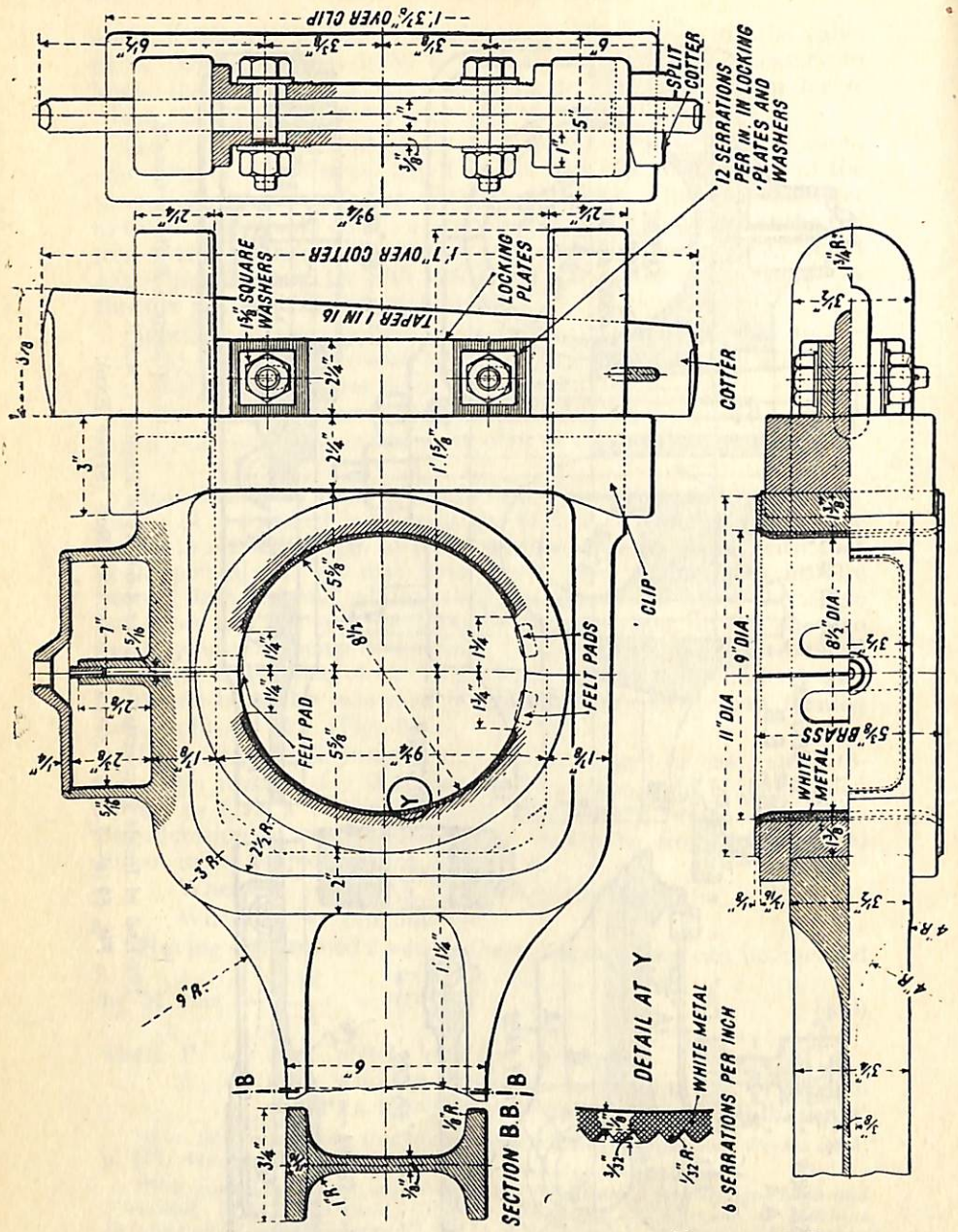
[15]For consideration of cottered joints see *Steam Loco. Design Data and Formulae*, E. A. Phillipson, Chapter IX. Also *A Manual of Machine Drawing and Design*, Chapter VI, by D. A. Low and A. W. Bevis. Longmans Green & Co., Ltd.



[*Courtesy Railway Gazette.*

Big-End of Inside Connecting Rod for 4-6-0 Type Engine, L.M.S.R.

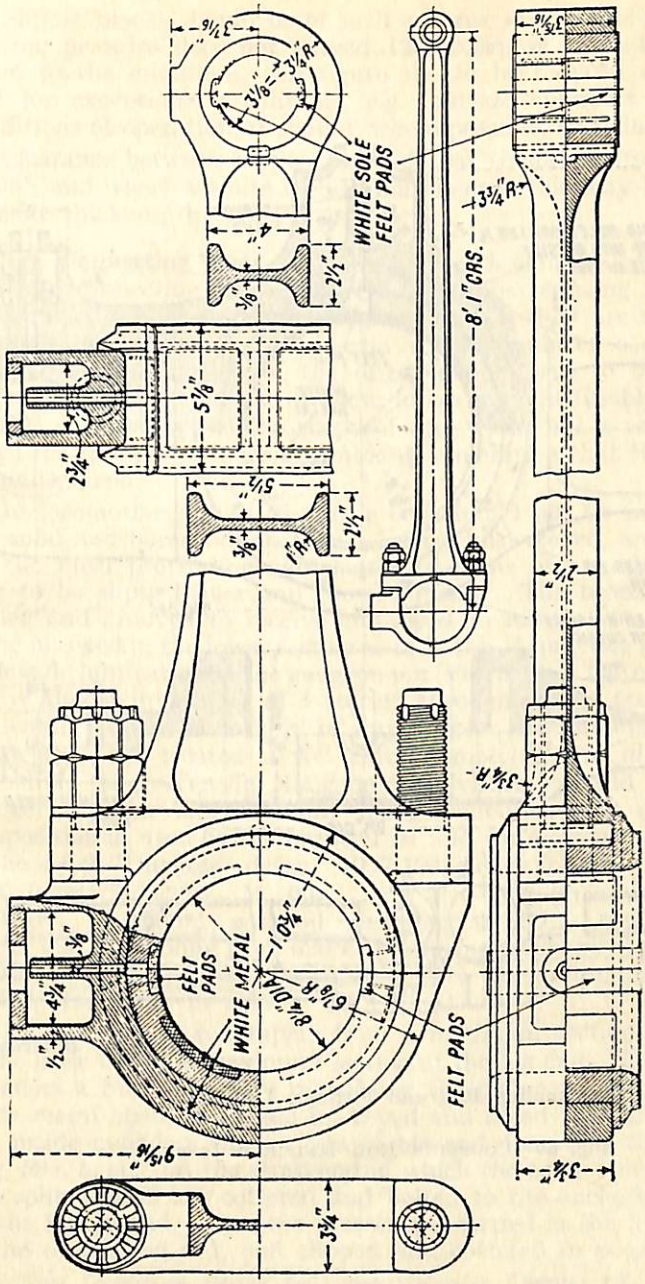
Fig. 59 (a)—L.M.S. Inside Connecting Rod (Strap End).



[*Courtesy Railway Gazette.*

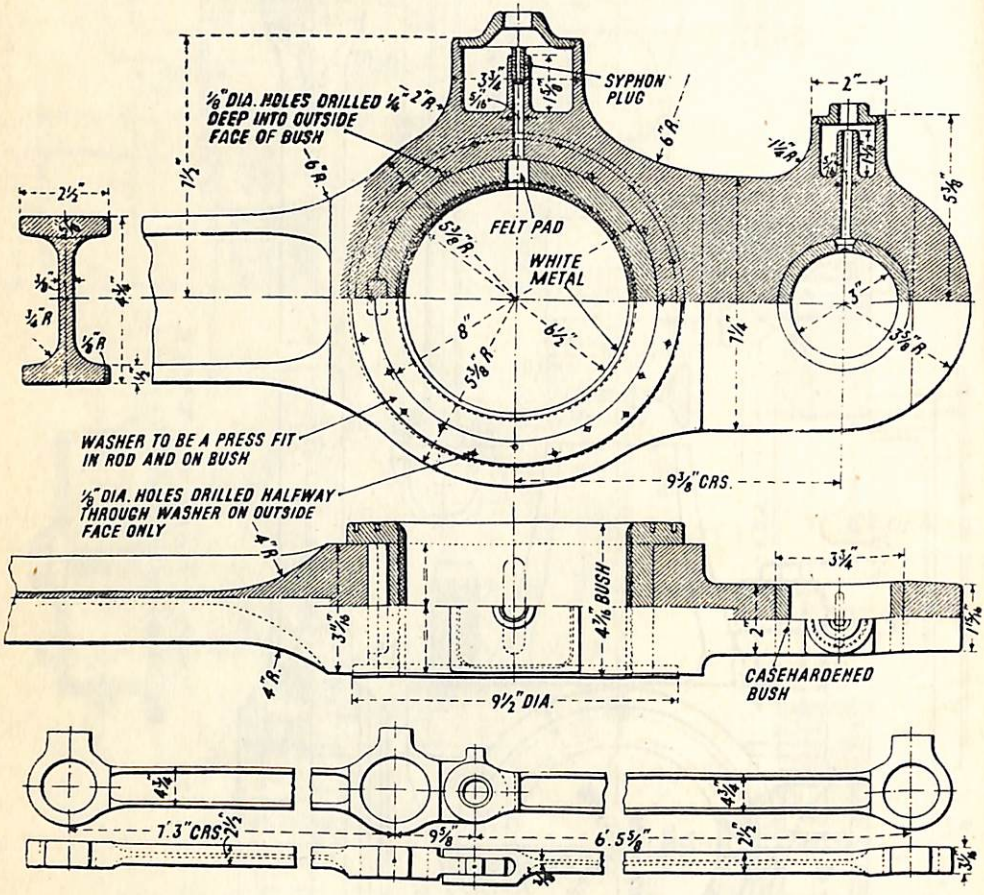
Inside Connecting Rod Big End. L.M.S.R.

Fig. 59 (b)—FORKED END.



Inside Connecting Rod, L.N.E.R. [*Courtesy Railway Gazette.*

Fig. 59 (c)—Marine Type. Big-End.


⅛" DIA. HOLES DRILLED ¼" DEEP INTO OUTSIDE FACE OF BUSH
SYPHON PLUG
FELT PAD
WHITE METAL
WASHER TO BE A PRESS FIT IN ROD AND ON BUSH
⅛" DIA. HOLES DRILLED HALFWAY THROUGH WASHER ON OUTSIDE FACE ONLY
9¾" CRS.
4⅞" BUSH
CASEHARDENED BUSH
9½" DIA.
7'3" CRS.
6'5⅝"

[*Courtesy Railway Gazette.*

Coupling Rod and Fittings, L.M.S.R.

Fig. 60—Coupling Rod Knuckle Joint.

Slipper blocks should be of such an area so that the maximum bearing pressure does not exceed 120 lb./sq. in. To keep wear down to the minimum, this figure should be reduced to 60 - 80, and for exceptional conditions, *e.g.*, outside cylinders in desert conditions of operation, it should, where possible, be reduced to 40.

Clearance between slipper and slide bar should be from 0·003"-0·005", and where a white metal lining is used this may be of convenient thickness up to $\frac{1}{8}$".

(iv) **Connecting Rods.**—In locomotive design the tendency is to make connecting rods relatively long thus reducing the effect of angularity. The greatest forces to which they are subjected, however, are those due to inertia which produces a deflection causing the application of the direct piston load to be slightly eccentric ; the derivation of these forces may justifiably be considered a little beyond the scope of this work, but a reference is given for cases where it is desirable or imperative that this should be undertaken.[16]

In locomotive design, the little end should not be forked, but left solid and bored out to take a bronze bush keyed, and pressed in, the bush protruding sufficiently to allow of an oil retaining ring to be slipped over and pressed home. The bronze bush is drilled and grooved to receive and assist in lubrication, felt pads being inserted in the lower sectors as in L.M.S.R. practice to provide underside lubrication to the gudgeon pin (Fig. 59*a*). The connecting rod is almost invariably of I section throughout the majority of its length, and is invariably of forged steel, the present practice is to use heat treated nickel chrome molybdenum alloy steel, H.T.M.M. (High Tensile Manganese Molybdenum) and "Vibrac" (Nickel Chrome Molybdenum) being widely used. A typical composition as used by the former L.M.S.R. for motion components in the 4-6-2 "Duchess" design (1937-1944) is as follows :—

C 0·3%, S_i 0·22%, M_n 0·6%, S & P 0·03% (max.), N_i 2·5%, C_r 0·66%, M_o 0·60%, with oil quenching at 850°C., tempering at 650°C. and air cooling from 620°C. producing mechanical properties of Breaking Stress 55-60 tons/sq. in. ; 25% elongation on 2 in. and 55-60% reduction in area.

The connecting rod tapers from a minimum section just rear of the little end to a maximum section at the big end. For outside cylinders a bush generally resembling the gudgeon pin bush, but white metal lined, is pressed in, keyed and fitted with an oil ring. For inside cylinders three types of big end commend themselves (Fig. 59*a*, *b*, *c*) ; (*a*) the strap end in which the connecting rod end and split brasses are cottered and bolted to the enclosing strap ; (*b*) the forked end, where the brasses are carried in the forked end of the connecting rod, and clipped and cottered in position, and

[16] *Steam Locomotive Design Data and Formulae*, Chapter IX.

(*c*) an adaptation of the marine type in which the big end is split vertically in two halves, and bolts up around the big end brass. Oil wells must be provided at each end, and oil ducts and (wick) trimmings provided (the former often being made from hand taps screwed in and twisted "off"), the whole being maintained dust proof either by screwed in hexagon headed plugs, or ordinary good quality bottle corks.

In the outside connecting rod with its pressed-in bush, no adjustment for wear is possible ; when the scrapping size has been reached, the bush is pressed out, sent for re-metalling and a substitute pressed in. The metal used for the anti-friction lining (unless roller bearing ends are being used) may approximate to, or be identical with, that used in the axle boxes, that used on the former L.M.S.R. consisting of S_n 59%, S_b 9·5%, C_u 3·0%, P_b 28·5%, whilst that used on the Central Argentine Railway consists of S_n 83·0%, C_u 11·5%, S_b 5·5%, and has a crushing load of 10,820 lbs./sq. in. Bearing pressures for big end journals vary from 1,000-2,000 lbs./sq. in. of projected area average figures being 1,400 lb. (inside), 1,600 (outside) for engines using superheated steam.

(v) **Coupling Rods.**—The whole of the foregoing section where applicable goes for coupling rods. There is, however, a choice of section ; they may be I section (Fig. 60), rectangular section, or rectangular section milled out (fluted) on the outside. The bushes as before are pressed in, keyed, fitted with oil rings, and white metalled. Where more than two axles are being coupled knuckle joints are provided, in such a way that the driving wheels and one other pair are always rigidly coupled.

Coupling rods are subjected to a complexity of stresses, due to centrifugal force, bending, torque, inertia and strains produced by flexing when negotiating curves. Safe working stresses should be taken as about 6 tons per sq. in. It is usual to bore out the bearing centres equal to the axle centres, but on four-coupled engines this distance is often increased by $\frac{1}{64}$"-$\frac{1}{32}$" to allow for frame strains under loads, or frame expansion in the region of the firebox. The outside diameter of coupling rod boss around the bush is usually 1·7-1·9 × crank pin diameter. Similar provision for lubrication of crankpin journals as before should be made.

When on their crank pins, coupling rods are retained by large screw-on washers, there being usually no allowance for wear in the bushes. In some designs the leading crank pin brasses are carried in a strap with cotter adjustment.

Valve gear components are made of the same material, machined for lightness, bushed and lubricated in the same manner, but later refinements include grease nipple and gun lubrication, needle roller bearings in place of motion pins with collars, and self-aligning

ball race in the eccentric rod banjo end which receives the return crank pin on outside Walschaerts gear.

(vi) **Motion Pins** for fifty odd years have been made to a diameter of D/10 (D = piston diameter), which in combination with accurate machining, case hardening, and centreless grinding has given consistently trouble free results. These pins are always in double shear, their shearing stresses arising from inertia forces of the associated component, which if assumed to move with a simple harmonic motion (acceleration proportional to displacement from mid-position) gives maximum inertia forces at the ends of the stroke, where acceleration is a maximum. This can be assessed from the S.H.M. formula where Force $= \frac{W}{g}\omega^2 r$ (xiv)

where W = weight of reciprocating part in lb.
ω = angular velocity of driving wheels in radians/sec., *i.e.*, 2π N.R.P.M./60.
r = half the stroke of the component in feet.
g = acceleration due to gravity = 32·2 ft./sec./sec.

Thus with modern British express passenger locomotives travelling at high speeds the various pins in the valve gear will have to transmit accelerating forces of between 2,000 and 3,000 lb.[17]; projected bearing pressures should be kept below 1,000 lb. per sq. in. wherever possible. Return crank pins due to inevitable wear should be made well in excess of D/10, an increase of 50% in this instance not being regarded as exceptional.

(vii) **Reversing Mechanism.**—A cross shaft carrying a bell crank and lifting links, whereby the link or link die blocks are raised or lowered, is provided under the direct control of the driver. Where the parts to be lifted are light, and fine variations in cut off are not required, but quick reversal is (*e.g.*, shunting tank engines) a notched sector with a long vertical lever pivoted to the frame, and connected to the bell crank on the cross shaft by a bar (reach rod) will generally be found to be the best arrangement.

Where the parts to be lifted are heavy, and fine cut off variations required, some form of screw mechanism, either vertical (L.N.E.R. big engine practice) or more generally horizontal, must be resorted to, along with some kind of locking mechanism to hold the gear at the desired cut off. Further, a cut off indicator marked in percentages from full forward gear to full reverse gear, and indicating "mid-position," must be provided in some locality within the driver's vision. It is also usual to counterbalance the parts to be lifted by means of dead weights or by the extending of a stiff spring; but for locomotives of either large dimensions, or of

[17]For an interesting treatment of this topic see *Design of Valve Gear Joint Pins*, by Geo. W. McArd, A.M.I.Mech.E. *Railway Gazette*, 11/7/52.

advanced design, power operated reversing gears (steam or compressed air) have been in use for a great many years with complete satisfaction.

In bringing this investigation on valves and valve gears to a conclusion mention must be made of that locomotive bête-noire the anti-vacuum valve (Fig. 61). Anti-vacuum, cylinder by-pass, air relief, and snifting valves, have been fitted to locomotives to destroy cylinder vacuum when drifting from the days of the pioneers in one form or another. In the superheated locomotive they couple this function with that of opening upon closing of the regulator valve, to admit air to circulate through the superheater and thence exhausted by way of the cylinders, in which instance they are known as anti-vacuum valves, or air snifting valves. In G.W.R. and L.M.S.R. practice a separate air relief valve is fitted to each cylinder, which admits air and a mixture of air and steam to the cylinders when drifting (Fig. 62). However time honoured traditions are being swept away by the removal of superheater header anti-vacuum valves from all engines of the former Southern Railway, it having been long known by actual experience that if the reversing gear is moved to and set at 45% there is neither vacuum nor compressed smoke box gases present in the valve chests and cylinders when drifting, a very desirable condition which can be helped by the admission of a small quantity of steam. Should the reverser be moved to full forward gear a vacuum of 10-12 inches will be created, and if placed at mid-gear, gas pressure may well reach 120 lb./sq. in. back pressure, in addition to which undesirable carbonisation of the cylinder lubricating oil occurs. Unfortunately air relief valves (especially L.M.S. type air relief valves) "chatter" unduly at low speeds, causing the engine so fitted to be distinctively noisy when drifting.

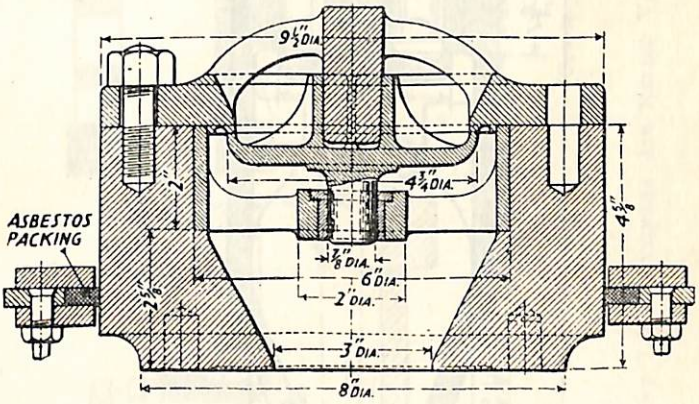
CHAPTER V.

LOCOMOTIVE DYNAMICS AND ASSOCIATED PARTS.

Some realisation of the nature of the forces set up when the locomotive is running would appear desirable if only on the grounds that such forces are set up and have to be accounted for in detailed design.

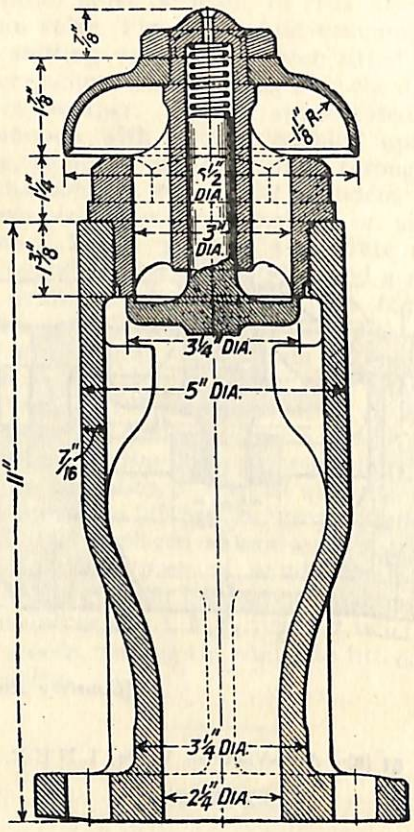
(1) Piston Velocity and Acceleration.

Differentiation must be made at the outset between piston speed (Chapter I (iii*a*)) which is the average speed of the piston and its velocity at any instant. The latter may easily be found



[*Courtesy Railway Gazette.*

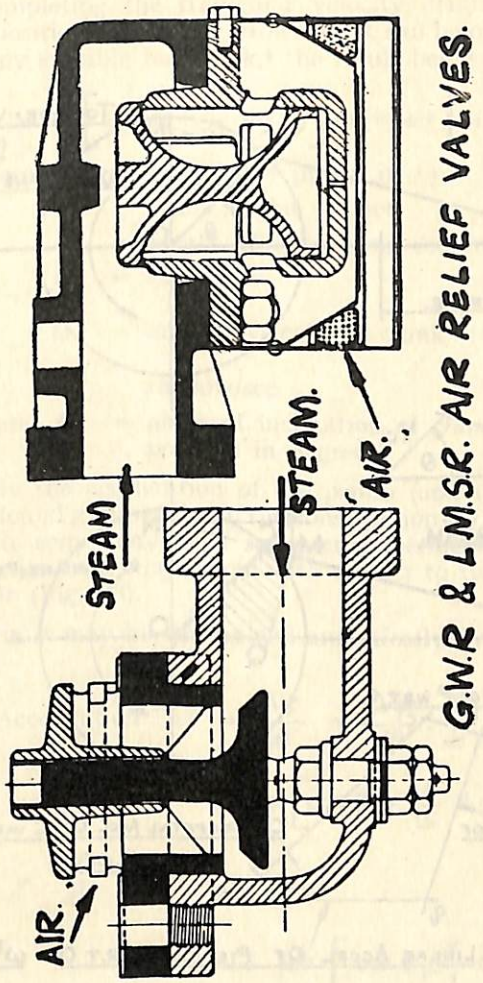
Fig. 61 (a)—Anti-Vacuum Valve, L.N.E.R.



[*Courtesy Railway Gazette.*

Fig. 61 (b)—Superheater Header Relief Valve, Southern Railway.

ANTI-VACUUM VALVES.



[*Courtesy British Railway:*

Fig. 62—G.W.R. and L.M.S. Cylinder Air Relief Valves.

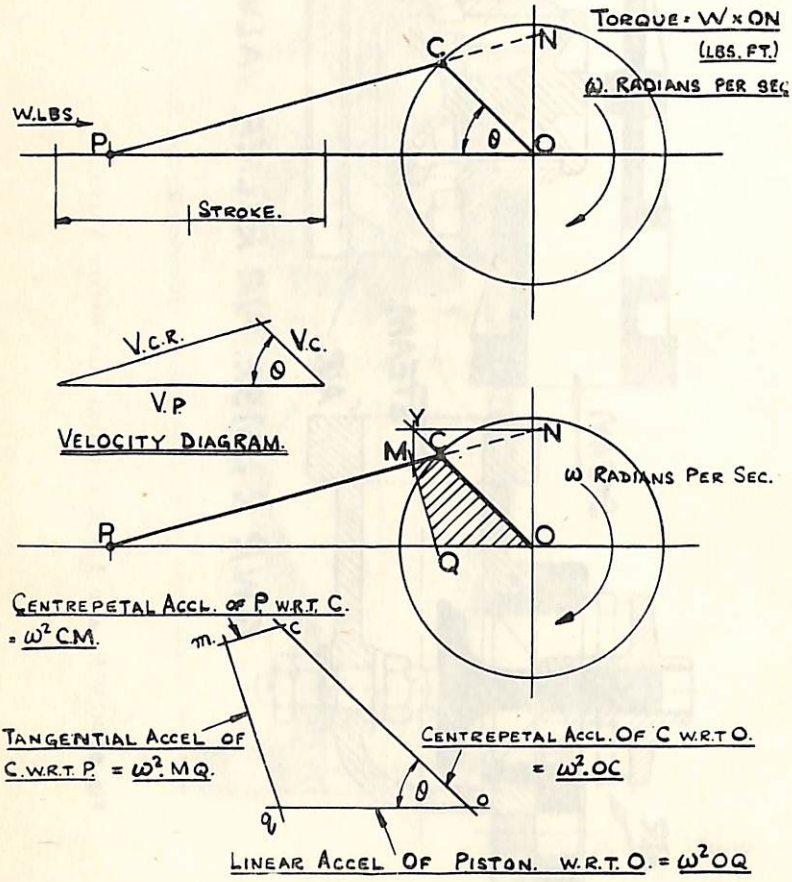


Fig. 63—Piston Acceleration Diagram. (Ritterhaus' Construction)

VELOCITY AND ACCELERATION DIAGRAMS.

graphically by drawing a velocity diagram to some convenient scale having first calculated the linear velocity of the crank pin and completing the triangular velocity diagram for the desired crank position (Fig. 63). However, it can be obtained analytically from any suitable text-book,[1] the result being :—

$$Vp = \omega r\left[\sin\theta + \frac{r}{2l}\sin 2\theta\right] \text{ or } = \omega r\left[\sin\theta + \frac{\sin 2\theta}{2n}\right] \quad \text{(i)}$$

Where Vp = velocity of piston in f.p.s.

r = crank radius in feet.

l = length of connecting rod in feet.

n = l/r.

ω = angular velocity of crank = $2\frac{\pi}{60}$ N.R.P.M. radians/sec.

and θ = angle of inclination of *crank* from dead centre position in degrees.

Similarly the acceleration of the piston (and its associated parts) can be found graphically by the constructions of Klein or Ritterhaus drawn to some convenient scale remembering that the centrepetal acceleration of the crank pin with respect to the centre of rotation is $= \omega^2 r$ (Fig. 63).

Again it may be determined analytically by the differentiation of (i) :—

$$\text{Acceleration } ``a'' = \frac{dVp}{dt} = \frac{dVp}{d\theta}\cdot\frac{d\theta}{dt}$$

$$= \omega r\left[\cos\theta + \frac{r}{2l}\,2\cos 2\theta\right]\frac{d\theta}{dt}$$

but $\omega = \dfrac{d\theta}{dt}$

$$= \mp\,\omega^2 r\left[\cos\theta \pm \frac{r}{l}\cos 2\theta\right] \quad \text{(ii)}$$

which is sufficiently accurate for all purposes provided a difference of sign is made when considering acceleration to be from front or back dead centre. As the locomotive is not considered to be a high speed engine, it is not usually necessary to use the Fourier Series for determining problems connected with acceleration of reciprocating parts. It can safely be assumed that the piston has no acceleration at or about the point at which the crank and

[1] See Bibliography.

connecting rod are at right angles, whence the piston due to effects of angularity will be at 0·45 × stroke furthest from the crank.

(2) Crankshaft Torque.

Is the turning moment on the crankshaft, being the product of the thrust or pull in the connecting rod × the radius of the crank, or more conveniently it is the product of the load on the piston × the intercept on the vertical axis of the crankshaft made by producing the connecting rod to cut the vertical axis (Fig. 63). It fluctuates from zero at the ends of the stroke to a maximum just prior to midstroke. Notwithstanding, however, a fly wheel need not be fitted as the driving wheels and their associated cranks and balance weights provide sufficient flywheel action to iron out the variations in torque.

(3) Locomotive Balancing.

By far the most important detailed investigation in locomotive design, is that of balancing for it largely determines the route availability of the locomotive and its harmful effects on the track over which it is designed to operate. There is a tendency to avoid the subject of balancing because of the technicalities involved though this is largely due to the failure to understand the underlying theory, which in reality is not particularly involved, so if only for reasons of completeness and importance it must be attempted.

At the outset it can be stated that *revolving* masses though they be in various planes, and at different angular positions in those planes can be completely balanced ; the various methods employed can all be found in suitable text books ; it is generally achieved in locomotive design where there are inside cranks, by placing the required balancing masses at crank pin radius opposite the revolving masses it is intended they should balance. Thus for a two-cylinder engine with inside cylinders, the outside crank pins and attendant revolving masses will be balanced by the inside balancing masses on the other side of the engine centre line. These balancing masses at crank pin radius take the form of inside crank web extensions, where there are no inside cranks, the revolving masses must be balanced in the wheel balance weights.

On the other hand *reciprocating* masses *cannot* be *completely* balanced, their disturbing effects can, however, be mitigated. Suppose a balance weight W_B (Fig. 64) be placed at crank radius r opposite the crank and equal to the total reciprocating weight W_R, the centrifugal force due to W_B has a component *in the line of stroke* $= \frac{W_B}{g}\omega^2 r \cos\theta$, which neutralises the unbalanced reciprocating force, but the component of the centrifugal force due to

W_B *perpendicular* to the line of stroke remains unbalanced and is equal to $\frac{W_B}{g} \omega^2 r \sin \theta$. Thus all that has been achieved is to introduce a further force which is zero at the ends of the stroke and is a maximum at mid-stroke (sine 90° = 1) and perpendicular to the original unbalanced force, and equal to it (Fig. 64).

As previously stated the harmful effect (*i.e.*, the magnitude) of this vertical component of the centrifugal force due to W_B can be reduced by the simple expedient of reducing W_B itself in the first instance, thus if in Fig. 64 $W_B r = kW_R r$ (where k is less than unity).

$$\text{Force in line of stroke} = \frac{W_B}{g} \omega^2 r (1 - k) \cos \theta \qquad (a)$$

and

$$\text{Force perpendicular to line of stroke} = \frac{W_B}{g} \omega^2 rk \sin \theta \qquad (b)$$

So that if $k = \frac{2}{3}$, *i.e.*, only $\frac{2}{3}$ the total reciprocating weight is to be balanced the expression (*a*) becomes

$$\text{Force in line of stroke} = \tfrac{1}{3} \frac{W_B}{g} \omega^2 r \cos \theta \qquad (c)$$

and

$$\text{Force perpendicular to line of stroke} = \tfrac{2}{3} \frac{W_B}{g} \omega^2 r \sin \theta \qquad (d)$$

It is this quantity (*d*) that is known as "*Hammer Blow*" and though for the engine its total may be less than that for each individual crank, it must be a minimum consistent with other considerations which will be seen.

It may appear desirable to reduce the hammer blow by further reducing the amount of the reciprocating weight to be balanced, but in doing so the component in the line of stroke will be increased, which may produce unpleasant linear oscillations, and more important still will increase the magnitude of the *Swaying Couple* which will cause the engine to sway from side to side severely at the front end, being the product of the unbalanced force in the line of stroke, and the perpendicular distance between cylinder centre lines. Thus it is not difficult to see that in the case of two cylinder engines having outside cylinders this quantity becomes of considerable magnitude and importance, therefore the value of "k" should not be less than a half.

Investigations carried out by the former L.M.S.R. on three locomotives of the 4-6-0 class 5 Mixed Traffic type with 6′ 0″

diameter driving wheels and a static driving axle load of 17·9 tons upon well greased rails (and anchored) and "revved" up to the given equivalent speeds gave the following results.[2]

TABLE V.

BALANCING TESTS CLASS 5 M.T. LOCOS (former) L.M.S.R.

No.	"k"	Equivalent Speed m.p.h.	Hammer-blow, tons	Lift of Wheels off Rails	Front End Oscillation
1	$\frac{2}{3}$	103	30	$2\frac{1}{2}$"	Normal
2	$\frac{1}{2}$	103	23	$\frac{1}{2}$"	Increased
3	$\frac{1}{3}$	99	15	nil	Alarming

In locomotive practice the following are considered to be revolving masses and reciprocating masses respectively.

(i) **Revolving.**—Outside crank pins, portion of coupling rod, inside cranks and pins (where applicable), portion of connecting rod, outside crank webs (where applicable), return crank and big end of eccentric rod, inside eccentrics (where applicable).

(ii) **Reciprocating.**—Piston and rod assembly complete, cross-head complete with gudgeon pin, arm, link, and combination lever (where applicable), and portion of connecting rod. All are assumed to be balanced by some balancing mass $= k \frac{W_R}{g}$ placed at crank pin radius.

In the coupled locomotive there may be eight (two cylinder) nine (three cylinder) or ten (four cylinder) planes in which revolving masses will occur, the plane containing a required balance weight should be taken as a reference plane (zero moments in that plane) and the weights, forces and moments calculated and tabulated remembering that all moments to one or the other side of the reference plane chosen shall be considered throughout as negative moments, and drawn vectorially in the opposite direction in which they appear to be acting.

To determine the fraction of the connecting rod to be taken as (i) reciprocating; (ii) revolving, the centre of gravity can be found by resting the rod on knife edges at the pin centres, and measuring the reactions which are inversely proportional to their distance from the centre of gravity. Let the distance from the

[2]See *Locomotive Practice and Performance in the 20th Century*, by C. J. Allen, Chapter VIII, published by W. Heffer & Sons, Ltd.

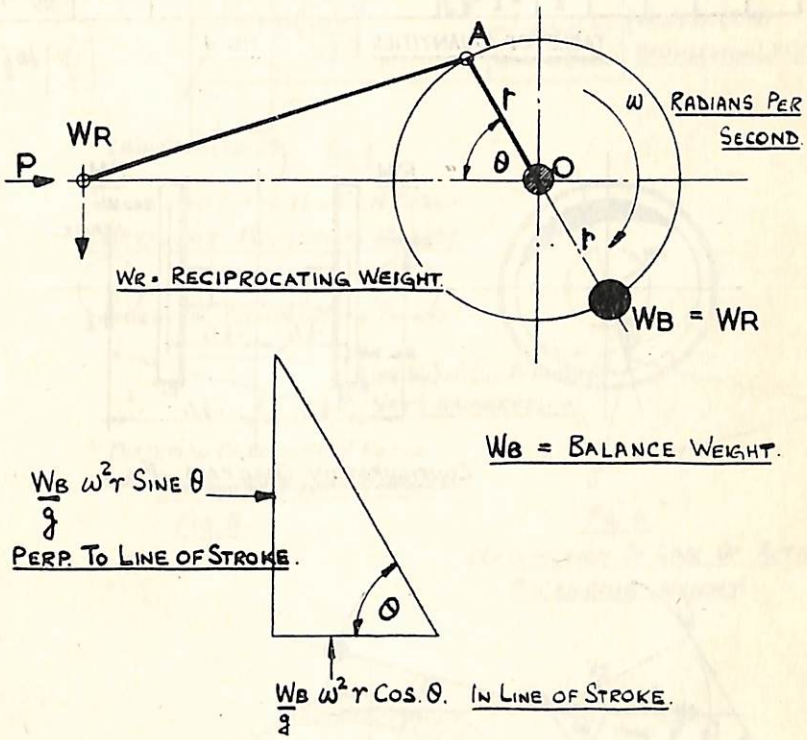
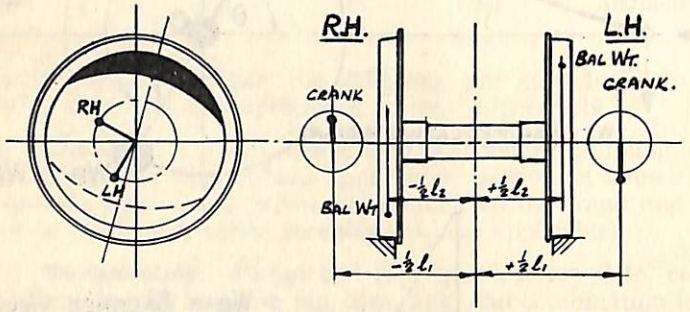


Fig. 64—Partial Balance of Reciprocating Masses.

CYL	W	r	PRIME FORCE W_r	l	PRIMARY MOMENT $W_r l$	WHEEL	L	BAL. MASS M	BAL MASS RADIUS R	MR	MRL
LH	W_A	r	$W_A r$	$l_1 - \frac{l_2}{2}$	$W_A r\left[l_1 - \frac{l_2}{2}\right]$	LH	0	M_A	R	$M_A R$	0
RH	W_B	r	$W_B r$	$-\frac{l_1 + l_2}{2}$	$-W_B r\left[\frac{l_1 + l_2}{2}\right]$	RH	l_2	M_B	R	$M_B R$	$M_B R l_2$

TABLE OF QUANTITIES FIG 1



CONFIGURATION DIAGRAM FIG 2

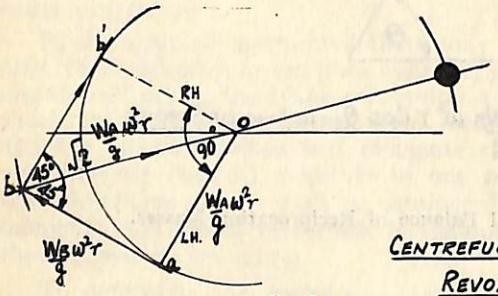


FIG 3

CENTREFUGAL FORCE DIAGRAM.
REVOLVING MASSES

Fig. 65—Graphical Representation of Locomotive Balancing.

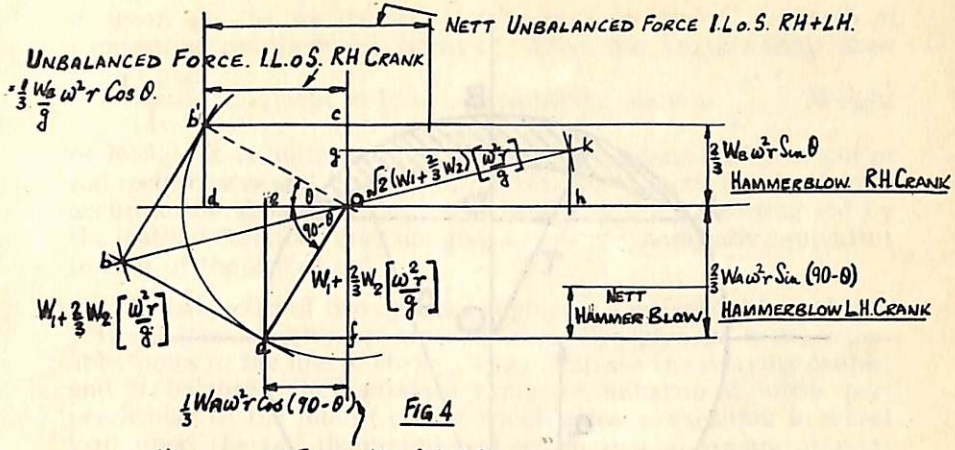
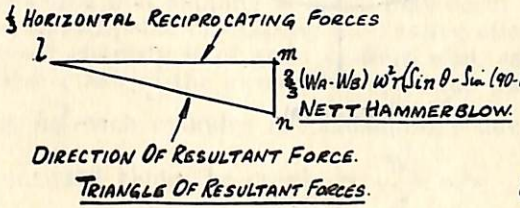


FIG 4

UNBALANCED FORCE I.L.o.S. L.H. CRANK.

EQUIVALENT REVOLVING MASSES.



TRIANGLE OF RESULTANT FORCES.

FIG 5.

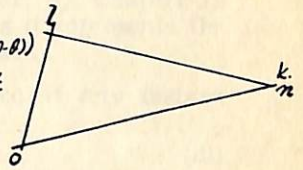


FIG 6

OL = DIRECTION OF LINE OF ACTION OF BALANCING WEIGHT.

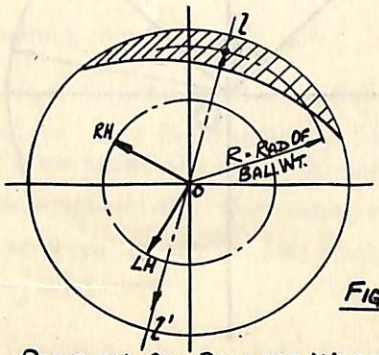


FIG 7

POSITION OF BALANCE WEIGHT.

Fig. 65—Graphical Representation of Locomotive Balancing.

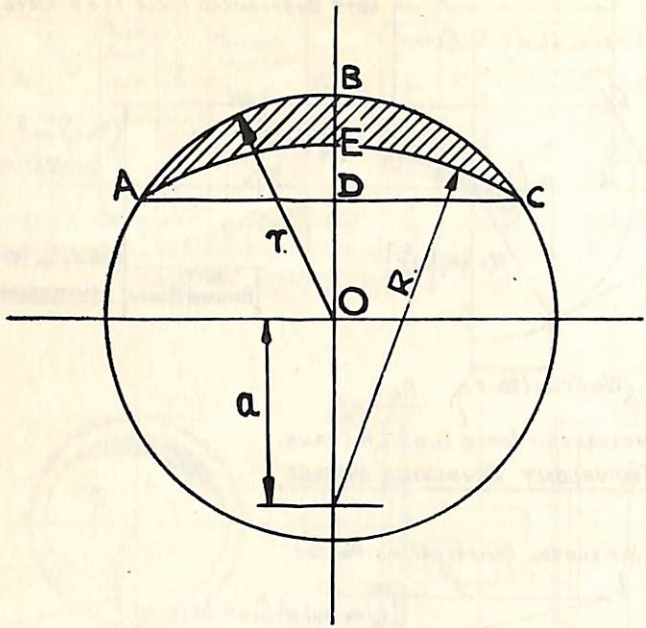


Fig. 66 (a).

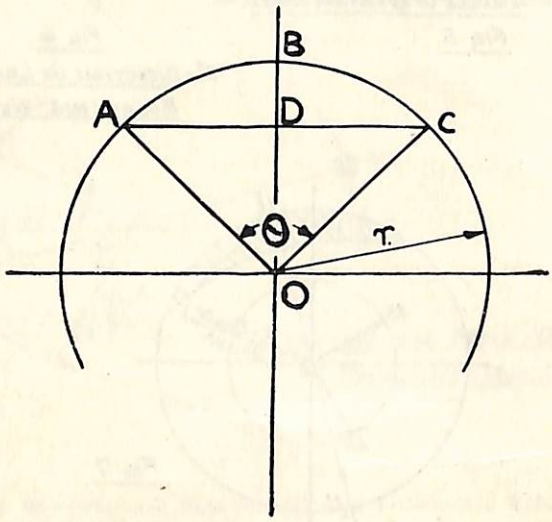


Fig. 66 (b).

gudgeon pin to the centre of gravity = $\bar{x}$, and L = length of connecting rod between centres. Then the reciprocating mass $= \frac{L - \bar{x}}{L} \times$ Weight of Rod, and revolving mass $= \frac{\bar{x}}{L} \times$ Weight of Rod. It is quite general, however, to assume that $\frac{1}{3}$ weight of rod reciprocates and $\frac{2}{3}$ weight of rod revolves ; further to be strictly accurate the above distribution of weight of the connecting rod by the method described does not give a system dynamically equivalent to that of the actual rod.[3]

The balancing of two cylinder engines is a considerable problem. If the reciprocating masses are unbalanced they give rise to objectionable forces in the line of stroke, which increase the swaying couple, and if balanced they produce excessive unbalanced forces perpendicular to the line of stroke which cause a variation in wheel load upon the rail (hammer-blow) which is a maximum at mid-stroke (theoretically, neglecting effects of angularity) which must alternately be added to and subtracted from the static wheel load. The latter value is of paramount importance as it may so diminish the wheel load that slipping at speed may occur and so the engine is rendered incapable of utilising its tractive effort (*cf.* Chapter I). The former quantity is of great concern also, as it represents the destructive effect of the locomotive upon the track.

Thus for each cylinder the unbalanced force at any instant acting outward along the crank, is $\frac{W_1}{g} \omega^2 r$ (iii)

where W_1 = total *Revolving* Weight and the resultant unbalanced force is $\sqrt{2} \frac{W_1}{g} \omega^2 r$ (iv)

in a direction bisecting the crank angles (Fig. 65).

The corresponding couple is $\frac{W_1}{g} \omega^2 r l_1$ (v)

l_1 = distance between cylinder centre lines.

Both (iv) and (v) may be completely balanced by suitable masses placed at some radius (R) in the driving wheels.

Let W_2 be the weight of the *reciprocating* masses per cylinder.

The unbalanced force for any angular position $\theta°$ of the crank is $\frac{W_2}{g} \omega^2 r \cos \theta$ (vi)

[3]See Articles 110-113, *Theory of Machines*. B. B. Low.

for one cylinder, and $\frac{W_2}{g}\ \omega^2 r \cos(\theta - 90°)$ for the other cylinder, giving a total unbalanced force at this position of

$$\frac{W_2}{g}\ \omega^2 r\ (\cos\theta + \sin\theta) \qquad \text{(vii)}$$

Let $y = (\cos\theta + \sin\theta)$ for a maximum value $\frac{dy}{d\theta} = 0$ so that $(-\sin\theta + \cos\theta) = 0 \quad \therefore \theta = 45°$ and $y = \sqrt{2}$

Hence (vii) is a maximum when $\theta = 45°$, 225° or 135°, and 315° and has a value of $\sqrt{2}\frac{W_2}{g}\ \omega^2 r$, the maximum unbalanced couple is $\frac{W_2}{g}\ \omega^2 r\, l_1$ and the maximum force at each crank is $\frac{W_2}{g}\ \omega^2 r$ acting horizontally.

If only $\frac{2}{3}$ W_2 is to be balanced, the *Equivalent Revolving Mass* $M = W_1 + \frac{2}{3} W_2$, and let this be balanced by an equivalent weight W_3 at crank pin radius (which is itself made up of weights W_R and W_T) which is added per cylinder at crank radius to balance the reciprocating and revolving masses respectively.

$$\text{Then } W_T = \left(\frac{W_1}{W_1 + \frac{2}{3} W_2}\right) \times W_3 \qquad \text{(viii)}$$

$$\text{and } W_R = \left(\frac{\frac{2}{3} W_2}{W_1 + \frac{2}{3} W_2}\right) \times W_3 \qquad \text{(viii}a\text{)}$$

Since W_T balances W_1 completely only the effects of W_2 and W_R remain, these may be resolved into three pairs of forces thus :—

(i) Horizontal Reciprocating Forces due to W_2.

(ii) Horizontal Components of Forces due to W_R.

(iii) Vertical Components of Forces due to W_R.

The horizontal components (ii) are balanced by $\frac{2}{3}$ of each reciprocating force leaving a final analysis of four unbalanced forces ; (i) vertical components of forces due to W_R, and (ii) $\frac{1}{3}$ of the horizontal reciprocating forces due to W_R. The former, as has been seen, is the cause of "hammerblow," and the latter the cause of the swaying couple. The actual variation in rail load is given by $W_s \mp \frac{W_R}{g}\ \omega^2 r$ per revolution of the driving wheel,

the negative sum being of great importance, which if itself be negative, *i.e.*, $\frac{W_R}{g}\ \omega^2 r$ is greater than W_s (static load) the wheel will leave the rail, the limiting condition being when $W_s = \frac{W_R}{g}\ \omega^2 r$.

The variation in tractive effort is due to the unbalanced effects of $\frac{1}{3}\ W_2$, the maximum horizontal unbalanced force

$$= \frac{\sqrt{2}}{3} \times \frac{W_2}{g}\ \omega^2 r \qquad \text{(ix)}$$

and is equal to the variation in tractive effort.

The maximum value of the swaying couple therefore is

$$\frac{1}{3}\ \frac{W_2}{g}\ \omega^2 r\, l_1 \qquad \text{(x)}$$

For complete balance of any engine, forces and couples due to (*a*) Revolving Masses; (*b*) Primary Forces; (*c*) Secondary Forces must balance among themselves, it has been stated that the first of these conditions can always be fulfilled, the second can usually be (but not in this instance), whilst the third rarely can. A peculiarity of the 90° two cylinder locomotive is that the secondary forces mutually balance, but for locomotives generally secondary balancing need not be undertaken.

It should be readily realised that 3 cylinder and 4 cylinder engines[4] are nearly self-balancing, especially the latter with adjacent reciprocating masses mutually opposed, in addition to which torque fluctuation and hammerblow are greatly reduced, which produces an all-round mechanically superior machine having a more gentle blast action upon the fire and requiring a lower factor of adhesion (Chapter I) which characteristics among others make it an attractive proposition.

It is not proposed to proceed further with the actual application of balancing to locomotives as various methods exist, useful references[5] are given herewith, it being left to the reader to decide which to him is the preferable method. Fig. 65 illustrates the graphical representation of the problem, the direction of the balancing (crescent shaped) masses being dependent on the numerical value of the various quantities W_1, W_2, etc. Having found the angular position of the centre of mass of the balancing mass, the actual balance weight is dependent upon the radius at which the centre

[4]Beyer-Garratt Locomotives must be considered here as two individual two cylinder locomotives.

[5]*Steam Locomotive Design Data and Formula.* E. A. Phillipson. *Loco. Engs. Pocket Book. Locomotive Counterbalancing* by Lawford H. Fry. *Rail Roads R.R.*, 56-2.

of mass is placed, usually as near to the wheel rim as possible, to reduce the weight of the mass, and its consequent hammerblow, whilst the crescent form is preferred in order to graduate its effect on the track.

The following formulae are offered in order to translate the balancing mass into crescent shapes. In Fig. 66*a* area of crescent ABCEA = (Area of segment ABCDA)—(area of segment AECDA).

Area of segment = Area of Sector OABCO—Area of Triangle OAC.

The area of a sector is given by $\frac{\theta}{360} \times \pi r^2$ (θ in degrees), and the distance of the centre of gravity of a segment from centre of the generating circle is given by $\left(\frac{\text{chord}^3}{12 \times \text{area}}\right)$. The balancing moment of crescent = Area of Segment AECDA × distance "*a*". Finally, the distance of the centre of mass of the crescent from the wheel centre 0 is as always = $\frac{\text{Moment of Crescent}}{\text{Area of Crescent}}$.

In all (coupled) locomotives the masses which partially balance the *reciprocating* masses may be distributed between the coupled wheels without upsetting the system dynamically, whilst reducing the hammerblow per wheel; the balancing mass which balances the *revolving* masses must however *be retained* within the wheel to which it refers. As a matter of practical interest it is interesting to note that the former G.W.R. "Hall" class 2 cylinder 4-6-0 locomotives having a 30" stroke have 70% of their reciprocating masses balanced, whilst the former Southern Railway "Merchant Navy" and "West Country" 3 cylinder 4-6-2 locomotives have no reciprocating balance whatever, the reasons for which and the consequent effects are left as an exercise in deduction for the reader (Fig. 67).

(4) **Wheels, Tyres and Axles.**

It is universal to use steel for the wheel centre, *i.e.*, hub, spokes and rim, the spoked wheel doubtless being a legacy of the first railway wheels which were wooden with iron rims. Open hearth acid steel of good resilient quality is specified with a typical analysis of C 0·26%, Si 0·21%, Mn 0·85%; S. & P. 0·033% maximum. The spokes should be tapered, eliptical in section, and any changes in section should be made with generous radii, in order to increase journal area, the spokes may slant outwards from rim to centre about 3"-4", thus forming a "dished" wheel. The crank web is cast integrally with the wheel centre and bored out to receive the crank pin. It is usual to provide one spoke for every 3"-4" of

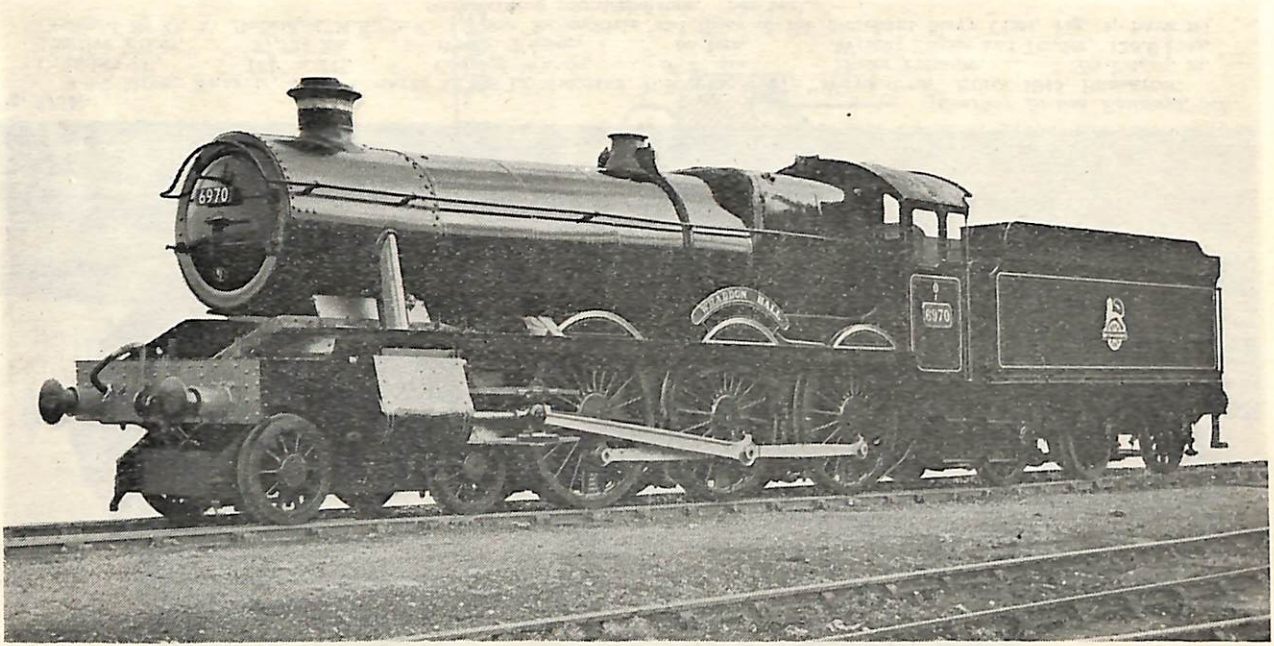


Fig. 67 (a). [*Courtesy British Railways, W.R.*

4-6-0 Mixed Traffic Hall Class Locomotive 6970, "Whaddon Hall."

Cylinders (2), ... 18½″ × 30″. Coupled Wheels, ... 6′ 0″ dia. Boiler Pressure ... 225 lbs./sq. in.
Tractive Effort, ... 27,275 lbs. Weight Engine, ... 75 tons. Weight Engine and Tender, 124.7 tons.
Photo shows "modified" Hall class with enlarged 21 element superheater introduced 1946 by C. Hawksworth, C.M.E., G.W.R.
These locomotives have 70% of reciprocating counterbalance. See text.

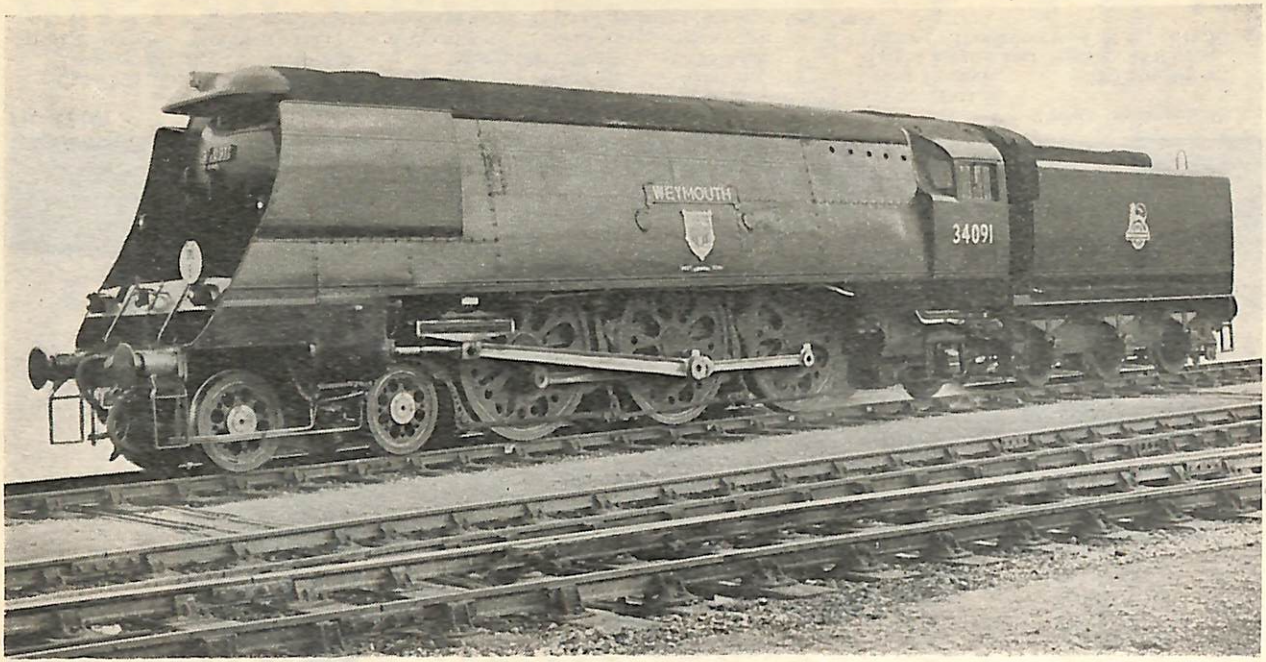


Fig. 67 (b). [*Courtesy British Railways, S.R.*

4-6-2 Mixed Traffic, West Country Class Locomotive, B.R. No. 34091, "Weymouth," Built 1945, Brighton.

Cylinders (3), ... 16½" × 24". Coupled Wheels, ... 6′ 2″ dia. Boiler Pressure, ... 250 lbs./sq. in.
Tractive Effort, ... 27,721 lbs. Weight Engine, ... 86 tons. Weight Engine and Tender, 128.6 tons.
Designed by O. V. Bulleid, C.M.E., S.R. These locomotives and those of the Merchant Navy Class, Fig. 4, have no reciprocating counterbalance. See text.

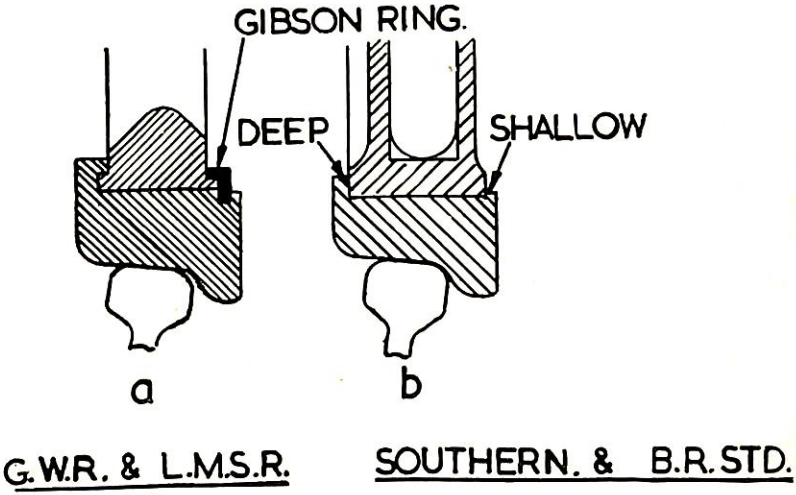
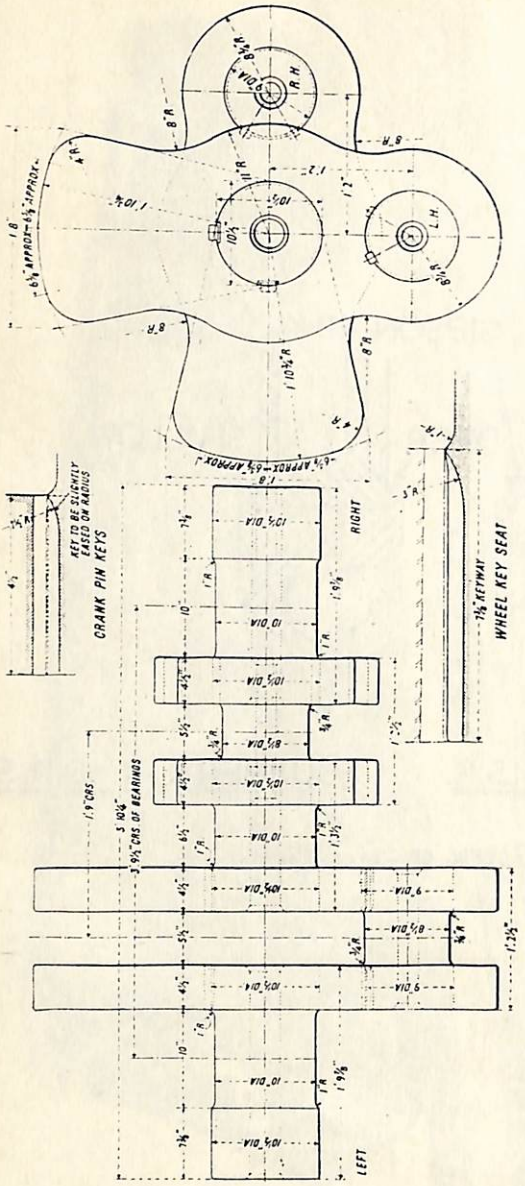


Fig. 68—Tyre Fastening



[*Courtesy Railway Gazette.*

Fig. 69—Built-up Balanced Crank-Axle, L.M.S.R.
(8P Duchess Class Locomotive).

wheel diameter, *e.g.*, the standard 6 ft. 8 in. L.N.E.R. wheel having 22 spokes gives 1 spoke per 3·65 ins. of diameter, and the L.M.S. 6 ft. 9 in. wheel having 21 spokes gives 1 spoke per 3·85 in. of diameter.

Box section wheels, commonly employed in the U.S.A., are used on former Southern Railway engines of O. V. Bulleid's design, and on the Ministry of Supply, W.D. 2-8-0 and 2-10-0 locomotives (Riddles design) whilst during the Webb & Whale[6] eras on the pre-grouping L.N.W.R. I section wheels were employed on freight locomotives, *e.g.*, L.N.W. 0-8-0 freight locomotives (later L.M.S. class G.1, G.2 and G.2A). It is claimed with justification that this more robust pattern of wheel is remarkably free from fractures in service.

Wheel boss sizes for coupled wheels are generally taken to be 2·0 to 2·5 times the axle diameter at the wheel seat, whilst for bogies, trucks, and other uncoupled wheels the boss size is taken to be twice the axle diameter at the wheel seat.

The wheel tyre (also of steel) is bored out to a smaller diameter than the wheel rim, heated and then shrunk on to the wheel, a common boring dimension being $D - \frac{D}{1000}$ (D = diameter of rim of wheel centre); whilst the formula employed by the erstwhile L.M.S.R. being $\left[\frac{D}{1200} + 0{\cdot}005''\right]$ to be subtracted from the diameter of the wheel rim, the shrinking allowance being 0·001″ per inch of tyre bore diameter. Thus a 60″ tyre must be finished internally between 59·993″ and 59·940″.

Tyres can be flush riveted to the rims in situ, though the rivets habitually work loose, and the holes are a source of weakness and fracture. It is preferable to employ the Gibson Ring method of fixture (Fig. 68*a*), whilst Fig. 68*b* illustrates the fixing employed on former Southern Railway engines of O. V. Bulleid's design, which method is now employed in British Railways standard locomotive practice. After shrinking on to the wheel centres, the tyres should be turned on the tread, to the finished diameter and requisite profile. Tyre thickness when new (for all other than narrow gauge engines) is from 3″-3½″ to allow of re-treading, scrapping taking place when the tyre thickness is down to 1½″-1⅝″.

On engines having more than three coupled axles, it is usual to fit either flangeless tyres, or tyres having thin section flanges on one or more pairs of the inner wheels, to allow of greater flexibility upon curved track.

[6]Francis William Webb (of compound notoriety), C.M.E., L.N.W.R., 1871-1903. George Whale, C.M.E., L.N.W.R., 1903-1909.

Axles are either of the straight or cranked variety, and should be made of high quality open hearth acid steel. There exists, however, a division of opinion as to whether the steel should be of low tensile strength (32-27 tons) or medium tensile strength (35-40 tons). The axle must be of sufficient diameter to allow of the requisite bearing pressure (Chapter II) and the section between journals should be of slightly reduced diameter to facilitate axle box removal.

The axle is pressed home into the machined-out bore of the wheel hub, and secured by one or more keys, whose key-way in the axle should be near the minimum required for secure holding, it being remembered that for all calculations the axle diameter excluding the key-ways must be used. Wherever possible a hollow axle is preferable, it being well known that weight for weight a hollow axle is stronger than a solid one.

Crank axles are frequently produced by casting, particularly if eccentric sheaves are to be mounted, though in modern practice with large locomotives, crank axles are built up; *i.e.*, crank webs with extensions are cut from steel slabs, machined, and keyed on to the half axles, crank pin turned and pressed home, and the crank axle then pressed into each wheel hub in turn (Fig. 69). Once again all changes in section, *i.e.*, from wheel to journal, axle to crank web, crank web to crank pin, should wherever possible be accompanied by generous radii.

(5) Brakes and Braking.

The requirements of braking are (*a*) to bring the locomotive and train rapidly and safely to a standstill in the event of an emergency application, or (*b*) to bring the engine and train smoothly to a gentle standstill under normal conditions of operation. To do this the applied braking force must be adequate for case (*a*) and capable of very fine graduation in case (*b*).

From first principles the Kinetic Energy of a moving body is $\frac{1}{2}$ Mass $\times$ Velocity^2, *i.e.*, $KE = \frac{WV^2}{2g}$ where W is the weight, and V is the velocity in feet per second and "*g*" the acceleration due to gravity. Thus if a train including engine of 500 tons is travelling at 60 m.p.h. (88 feet per sec.) it possesses a

$$KE = \frac{500 \times 88^2}{2 \times 32{\cdot}2} = 60{,}000 \text{ ft. tons}$$

and the average retarding force required to bring this train to rest (on a level track neglecting windage) in 1760 feet, *i.e.*, $\frac{1}{3}$ mile is $\frac{60{,}000}{1760} = 34$ tons (a condition not greatly divorced from reality

as obtaining on some short block sections on our home railways) whilst the theoretical braking force available is given by

$$\frac{P \times A \times L}{2240} \text{ tons} \qquad \text{(xi)}$$

Where A = Area of brake piston or pistons.
P = Available pressure in brake cylinder(s) lb./sq. in.
L = Ratio of brake rigging leverage.

It is general also to add up to 10% of the theoretical Kinetic Energy obtained just previously, in British practice to account for the rotational energy of the wheels of the train and the revolving masses of the engine.

Like tractive effort, the maximum braking force that can be utilised is a function of both adhesive weight and the co-efficent, of friction between wheel and rail, it being general in discussing braking to refer to the "braking efficiency" (particularly) of a train which is obtained from

$$\text{Braking Efficiency } \% = \frac{3{\cdot}34 \text{ (V m.p.h.)}^2}{\text{braking distance in feet}} \qquad \text{(xii)}$$

In order to avoid physical discomfort, excessive brake block wear, etc., it is general to aim at a retardation of three to four feet per second, per second. It should be remembered the braking force required varies directly as the square of the speed; whilst the leverage in the braking system should not greatly exceed 10 : 1 (about 7 : 1 for steam brakes) otherwise a great deal of ineffective travel of the piston occurs upon brake block wear.

Brake blocks should be of soft cast iron and easily renewable, whilst the rigging should be arranged to fulfil the following requisite features :—

(i) To provide adequate braking surface (*i.e.*, two blocks per wheel if necessary).

(ii) The line of action of braking force should be radially inward to the wheel centre of application.

(iii) The blocks to be centred a little above or below the centre line of the wheel of application.

(iv) The hangers should "pull" on, and when released should be gravity aided to their position of rest (*i.e.*, the hangars should be slightly inclined to the vertical when hard on).

(v) Brake force at each block to be identical (as far as possible for all wheels equally loaded), and where twin blocks per wheel are used the force in each block to be equal.

(vi) The components of the braking system should be in tension when applying the braking force.

(vii) Twin pull rods (in place of a central one) are preferable as the resulting bending moment in the cross beams is far smaller, though greater care is required in making brake adjustments to get equal pull. A turn buckle is generally preferable to multi-holed rod ends as a finer adjustment can be made, and the holes themselves are a source of weakness.

(viii) Where a hand brake is used, in conjunction with a power brake, its slotted link should pull on the brake piston cross-head or brake shaft, so that equal movement of all the rigging occurs.

(ix) The locomotive coupled wheels and all tender wheels should be braked, provision being made to obtain a braking force of 55-65% of the adhesive weight and 80% of tender weight.

The braking of adjacent wheels by one brake hanger centrally placed and a toggle mechanism is often discouraged on the grounds of setting up dangerous stresses in the coupling rods, However, Fig. 70*a* shows this type of braking as applied to the former Southern Railway Pacifics, without adverse effects or criticism.

Calculations in braking forces are usually a matter of resolving moments, it being possible to employ configurations in which any of the three orders of levers obtain. Fig. 70*b* shows the general disposition of brake gear as applied to locomotives of British design practice (uncompensated) whilst for fuller treatment of the subject the reader is referred to the appropriate bibliography.[7]

CHAPTER VI.

LOCOMOTIVE ACCESSORIES, ESSENTIAL FITTINGS AND REFINEMENTS.

Heretofore attention has been concentrated on the locomotive from various aspects. It must now be regarded as a corporate entity awaiting the addition of many important accessories without some of which it would not function, whilst the addition of others make it a far more attractive creation than it has often been in the past without them.

In the early days, in fact up to about forty years ago, the majority of the fittings to be described were manufactured by the

[7] *Steam Locomotive Design Data and Formulae*, by E. A. Phillipson, A.M.I.Mech.E. *Confines of Braking*, I-V *Railway Gazette*, October-November, 1952. Also literature available on request from The Westinghouse Brake and Signal Co., Ltd., and The Vacuum Brake Co., Ltd.

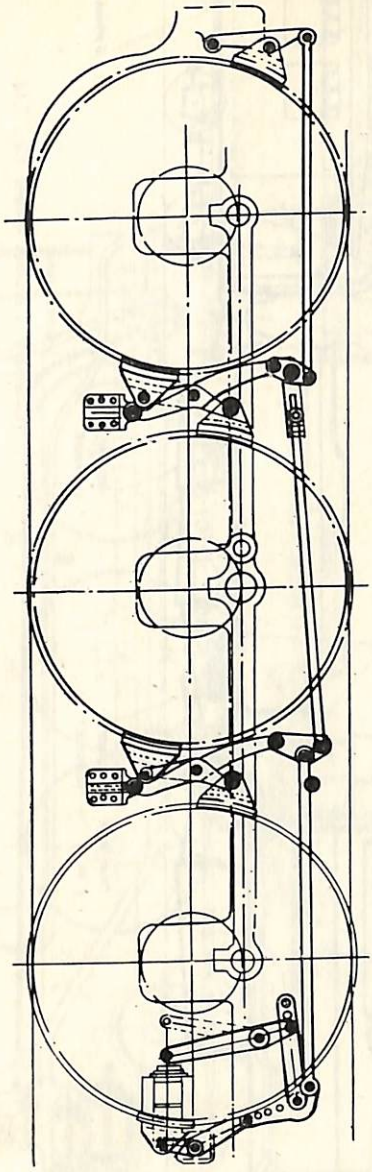
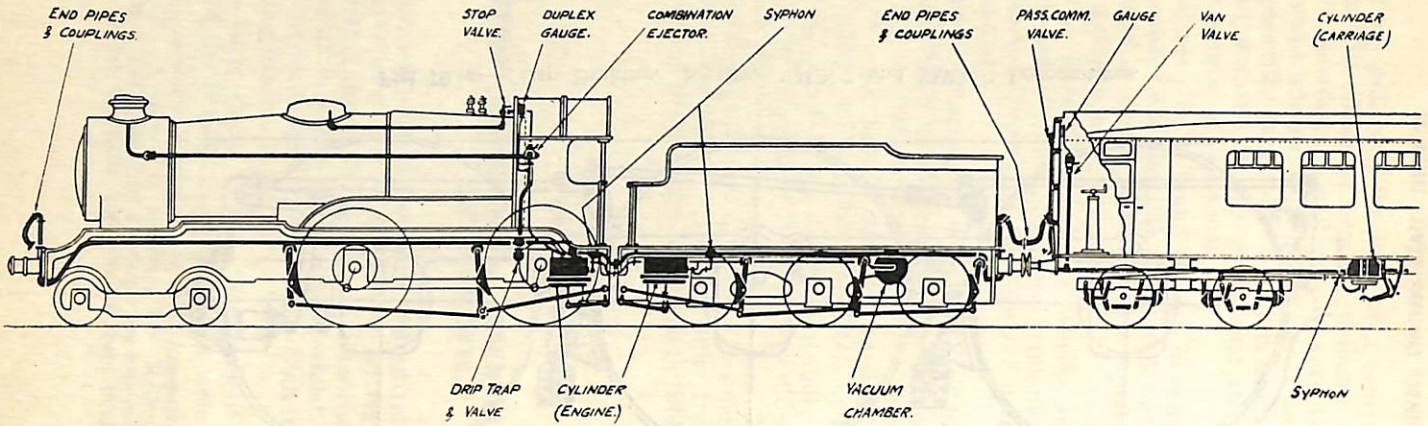
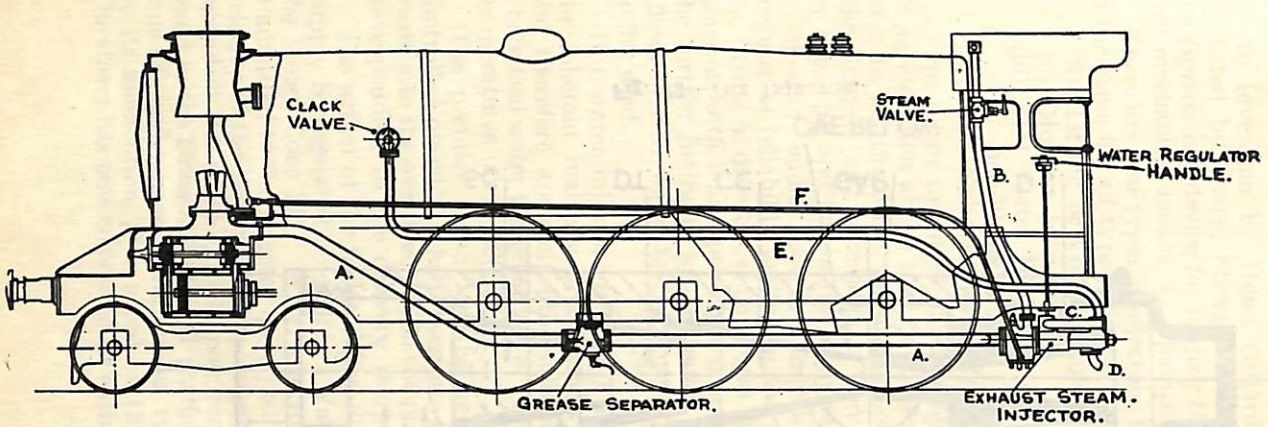


Fig. 70 (a)—Clasp Braking. S. Rly. "M.N." and "W.C." Locomotives.



[*Courtesy Vacuum Brake Co., Ltd.*

Fig. 70 (b)—General Arrangement of Vacuum Automatic Brake on Engine and Tender and Bogie Carriage with Guard's Compartment.



FIXING ARRANGEMENT OF
METCALFE'S PATENT EXHAUST STEAM INJECTOR—CLASS "J."
(AS FITTED TO S.R 'LORD NELSON' CLASS.)

REF.	PIPE.
A.	EXHAUST STEAM PIPE.
B.	LIVE STEAM PIPE.
C.	FEED WATER PIPE.
D.	OVERFLOW PIPE.
E.	DELIVERY PIPE.
F.	CONNECTION TO ENGINE S' PIPE.

Fig. 71.

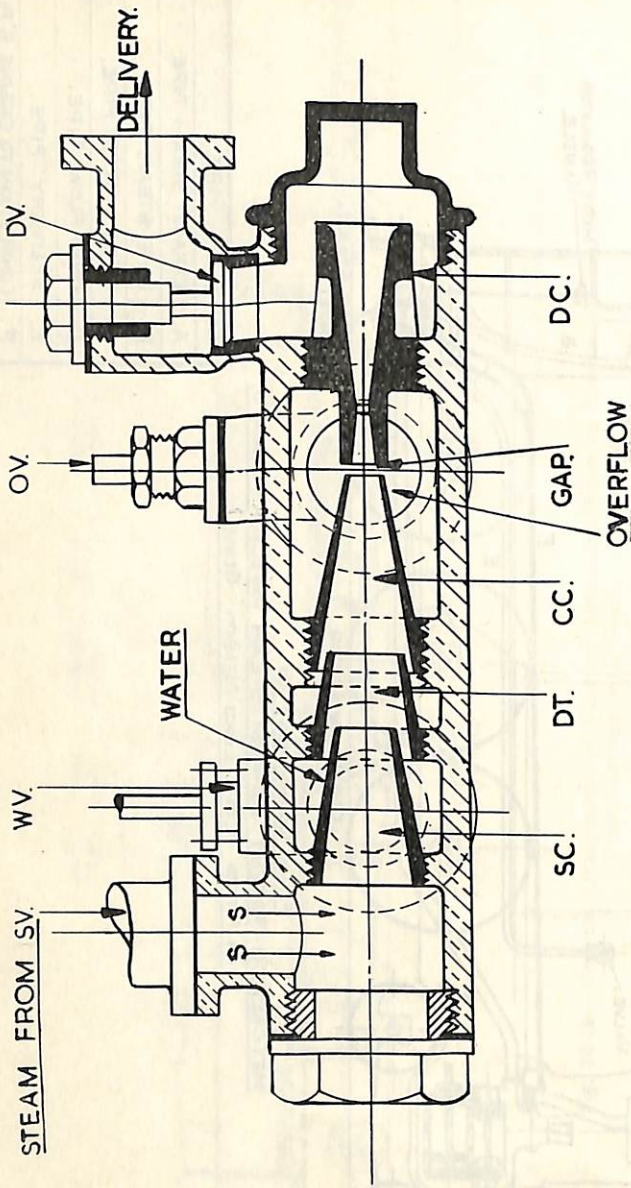


Fig. 72—The Injector.

locomotive builders, but have since become the speciality of a number of associated firms.

(1) Injectors.

An invention by Henri Giffard in 1859, and patented and marketed by Sharp, Stewart & Co., Ltd. (Manchester) in 1860, the *Injector* is a device for feeding the water into the boiler without any mechanical means, *i.e.*, pumps.

There are four types of injectors, though the underlying principle is the same for all types, viz. :—

(i) Lifting or Combination
(ii) Monitor (non-lifting)
(iii) Hot Water Injectors (lifting and non-lifting)

} Working off live steam.

(iv) The Exhaust Steam Injector.

Type (i) are mounted direct upon the boiler (firebox) backplate at an elevation above that of the water supply and therefore first have to lift the water up to the point of delivery. Despite their compactness and accessibility they have been largely displaced by types (ii) and (iii) in the live steam group.

Monitor under footplate injectors (type (ii)), have special features of reliability and automatic starting and restarting, whilst the fitting of different cone ranges enables them to be used in conjunction with boiler pressures in excess of 250 lb. per sq. in., and the further fitting of an automatic loaded type of overflow valve allows of feed water temperatures up to 130°F. into the injector, to be used.

Hot water injectors type (iii) have been evolved to overcome the inherent inability of injectors to handle heated feed water, and therefore are eminently suitable for service in hot climates where feed water temperatures of 140°F. can easily result in delivery temperatures up to 260°F.

The peculiar feature of the hot water injector is a pressure controlled overflow which prevents the water jet from flooding to waste into the overflow, due to the pressure in the overflow chamber corresponding to the delivery temperature.

The action of the injector (see Fig. 72), is such that the Steam Supply S, regulated by a Steam Valve SV, is accelerated through the Steam Cone SC. Water controlled by the Water Valve WV is admitted to the injector body just ahead of the SC orifice. The velocity of the steam at this point should be around 1,700 feet per second (1,160 m.p.h.), and upon coming in contact with the feed water in the Draught Tube DT partially condenses in the combining cone CC, and carries forward about twelve times its weight of water by its momentum, at a speed of 131 feet per second (90 m.p.h.). The steam has now converted its entire pressure energy into velocity

and is therefore able to carry the resulting stream across the gap G into the convergent divergent delivery cone DC from whence the water is fed to the boiler clack valve past the delivery valve DV.

Should the water or steam supply be interrupted, or alternatively either or both be excessive, it will flood to waste via the overflow valve OV. Once in the delivery cone DC the stream converts a part of its kinetic energy back into pressure energy sufficient to force itself into the boiler against the boiler pressure, at a delivery pressure some 50-80 lb. per sq. in. above the boiler pressure. The water fed to all injectors should be free from suspended foreign bodies, to which end, strainers should be placed between tanks and the injector.

The exhaust steam injector, operates on the same principle as the live steam injector, viz., the utilisation of the heat and velocity obtained from the expansion of steam and the imparting of them to water, though the cone diameters are altered to utilise low pressure steam diverted from the engine blast pipe. As the working of the exhaust injector depends on the fact that exhaust steam flowing into a vacuum attains a high velocity the cones are so designed that at the point of contact of steam and water the condensation of the steam shall be as efficient as possible thus obtaining a high vacuum.

The exhaust steam injector (Fig. 73), of which the Davies & Metcalfe type is very widely used on locomotives throughout the world, operates as a feed water heater so long as the engine is working. Upon closing of the regulator valve, it automatically changes over to live steam working, and back again to exhaust steam working upon re-opening of the regulator, the change over being controlled by steam pressure in the valve chest acting upon an automatic valve which shuts off the auxiliary live steam supply when exhaust steam becomes available. A small supplement of live steam is used and is controlled by the exhaust steam jet, for the purpose of increasing the delivery pressure up to present day boiler pressures.

In its present form the Davies & Metcalfe "Class J" exhaust steam injector represents the latest development of this product, and has a number of desirable features over and above its predecessors, among which is simplification of the automatic control system, the substitution of a manually operated water valve for a water valve controlled by live steam thus making its operation identical at all times with that of a live steam under foot plate non-lifting injector, and the addition of an automatic choke which improves feeding at low pressures, thus assisting greatly shed staff.

Savings of up to 10% in coal and water consumption are regularly maintained in service due to the fact that the exhaust steam heats the feed water and forces it into the boiler.

There exists a wide range of injectors for all conditions of service. The principal manufacturers Davies & Metcalfe, Ltd., and Gresham & Craven, Ltd., should be consulted before finally deciding on which type should be used, it being general, however, to employ an exhaust steam injector for continuous working, whilst one or two live steam injectors are carried in addition for rapid filling up of the boiler when so desired.

The *Grease Separator* is an indispensable fitting used in conjunction with the exhaust steam injector, and consists of a cast spherical chamber with a flanged inlet and outlet. The incoming steam is given a spiral twist by a helical vane, thus the heavier water and oil particles are flung radially outward and drain down the walls of the chamber into the sump below.

A further baffle plate just ahead of the helical vane and in front of the outlet, ensures the prevention of unwanted particles of ash, etc., passing to the injector. In the sump an automatic drip valve opens under the pressure of the exhaust steam thus draining the interior of the separator whilst the engine is using boiler steam (Fig. 74).

The Grease Separator must always be placed in the exhaust steam line between the blast pipe and the injector (Fig. 71).

Boiler feed water after leaving the injector is admitted to the boiler through *Clack Valves*, mounted usually on the first ring of the boiler barrel, or alternatively on the barrel top when it is usual to employ a "top feed" method of introducing the water to the boiler. In this instance water after passing the clack valve is delivered in the form of a finely divided spray, cascading down a series of internal trays placed above the main steam pipe thus mingling with the steam in the steam space before coming in contact with the boiler water (Fig. 75*a*).

A combined clack valve and water heater manufactured by Gresham & Craven, bolting directly on to boiler barrel, giving a water discharge temperature within 50°F. of the boiler steam temperature, is both compact and relatively simple in design (Fig. 75*b*), and fulfils adequately the functions of the top feed apparatus provided the boiler feed water is relatively free from alkalinity and suspended impurities.

(2) **Brake Ejectors.**

Locomotives intended for use with rolling stock fitted with the continuous automatic vacuum brake must be equipped with a vacuum brake ejector. This apparatus consists basically of two cones housed in a suitable casting supplied independently with boiler steam which issues with high velocity from them in such a manner that evacuation of the train-pipe, vacuum brake cylinders, reservoirs, etc., of all air takes place. The exhaust

steam is led along a large bore pipe (about 2 in.) alongside the boiler, and thence through an elbow into the smoke box and up the chimney (Fig. 70*b* and 76*b*).

The small ejector (small cone) is used for continuous running conditions in place of the now almost obsolete vacuum pump,[1] whilst the large ejector is used (*i.e.*, the large cone) for "blowing up" rapidly the full vacuum after a brake application.

Though 21 in. is the Board of Trade figure, ejectors are available (notably the new Davies & Metcalfe R.M. Vacuum Brake Ejector) which are capable of creating 27 in. (maximum) of vacuum thus providing both adequate vacuum raising powers for the longest of trains, and an ability to counteract the "leaking on" of brakes due to train leakage (Fig. 76).

Closely associated with the ejector is the driver's brake valve whereby a controlled quantity of air can be admitted into the braking system which then is isolated from the small ejector (which in turn is in connection with the vacuum reservoirs only), thus gradually or otherwise applying the brakes. Upon returning the driver's brake valve to the "*Off*" or to the "*Running*" position the large ejector is brought into action.

If a steam brake is to be used on the engine (and tender) an associated steam brake valve is incorporated with the ejector whereby the application of the steam brake is controlled by, and is in direct proportion to, the application of the vacuum brake. When working uncoupled, such locomotives can be braked by the continued use of the vacuum brake mechanism or they can be braked by the direct manual application of the steam brake handle which then is usually trigger released from the dual control for this purpose (Fig. 76*a*).

Brake pistons and cylinders for use with the automatic vacuum brake are of necessity large in diameter (about 20 in.) as it will be appreciated that the pressure applying medium is the atmosphere having a maximum pressure of 14·69 lb. per sq. in., at sea level.

Modern trends in fitting vacuum brake gear call for the placing of all steam components outside the cab, viz., steam stop valve, steam valves for large and small cones, and the ejector body, whilst the air components, viz., the driver's brake valve (with or without the automatic steam brake valve), vacuum gauges, etc., are retained inside the cab, there being advantages for this trend undoubtedly though it means duplication in the number of separate fittings required, location, fitting, and interconnecting piping and rodding.

Once again the principal manufacturers of vacuum brake equipment should be consulted before final choice of apparatus is made ;

[1]Engines of the former G.W.R. still retain the Vacuum Pump.

the principal manufacturers and some of their specialities are given below :—

(i) The Vacuum Brake Co., Ltd. (Gresham & Craven), manufacturers of Vacuum Brake Cylinders, Brake Slack Adjusters, S.J. and Dreadnought Combination Ejectors, S.S.J. Ejector, Duplex Stop Valve, and Drivers Brake Valve.

(ii) Davies & Metcalfe, Ltd., Manufacturers of the H.T. Vacuum Brake Ejector, R.M. Ejector, and Automatic Steam Brake Valve, etc.

For full information on both brake ejectors and injectors, the reader is referred to illustrated brochures supplied on request by both manufacturers.

(3) **Westinghouse Automatic Air Brake** (Fig. 78).

In this brake system air compressed by a steam air pump compressor up to a pressure of 120 lb. per sq. in. is admitted to the brake cylinders, the consequent movement of the pistons applying the brake. The compressed air is stored in a main reservoir on the engine, this is then fed by the driver's brake valve into the train pipe, and via the triple valves into the auxiliary reservoirs.

Application of the brake is occasioned by reducing the pressure in the train pipe causing the compressed air stored in the auxiliary reservoirs to flow to the brake cylinders, the brake is released by restoring the pressure in the train pipe, the triple valves moving to cut off the compressed air supply and opening the brake cylinders to atmosphere, whence powerful springs return the pistons to the release position.

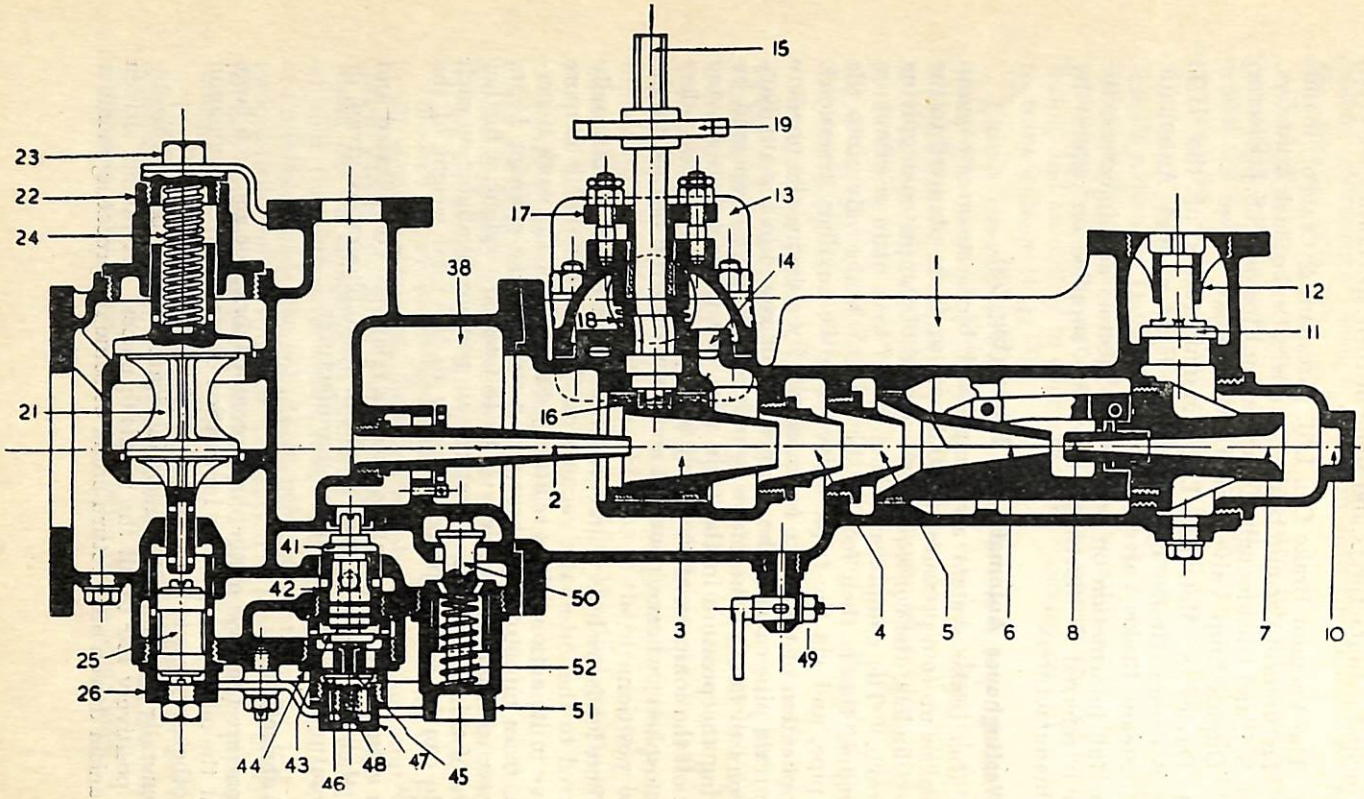
Where loads are heavy, and gradients severe this brake is usually preferred to the automatic vacuum on the grounds that it is more positive in its action ; it is relatively compact for the work it performs, train pipes of 1 in. bore and a single brake cylinder 14 in. diameter will control a vehicle up to 45 tons tare weight, a further feature of this system being that the air pressure available on each vehicle can be pre-set proportionately to the total weight of the vehicle.

In actual fact, this brake system is relatively complicated and its further description is best sought in the bibliography given, or in literature available from the manufacturer.

(4) **Lubricators.**

Gone are the days of the displacement lubricator and to a large extent the oil box too, as the lubrication of locomotives in common with other services is too vital a matter to be left to chance.

Generally speaking mechanical lubricators are in vogue, whereby oil is positively fed to all the requisite places. A ratchet driven mechanism driven off a return crank or some reciprocating member



Ref.	*Part.*
1	Injector Casing.
2-8	Injector Cones.
11-12	Back Pressure Valve and Guide.
13-19	Water Valve and Regulator.
21-26	Exhaust Steam Admission Control.
29-37	Overflow Valve and Automatic Control.
38	Exhaust Valve Casing.
41-45	Change-Over Control System.
46-48	Drio Valve.
50-52	Automatic Choke.

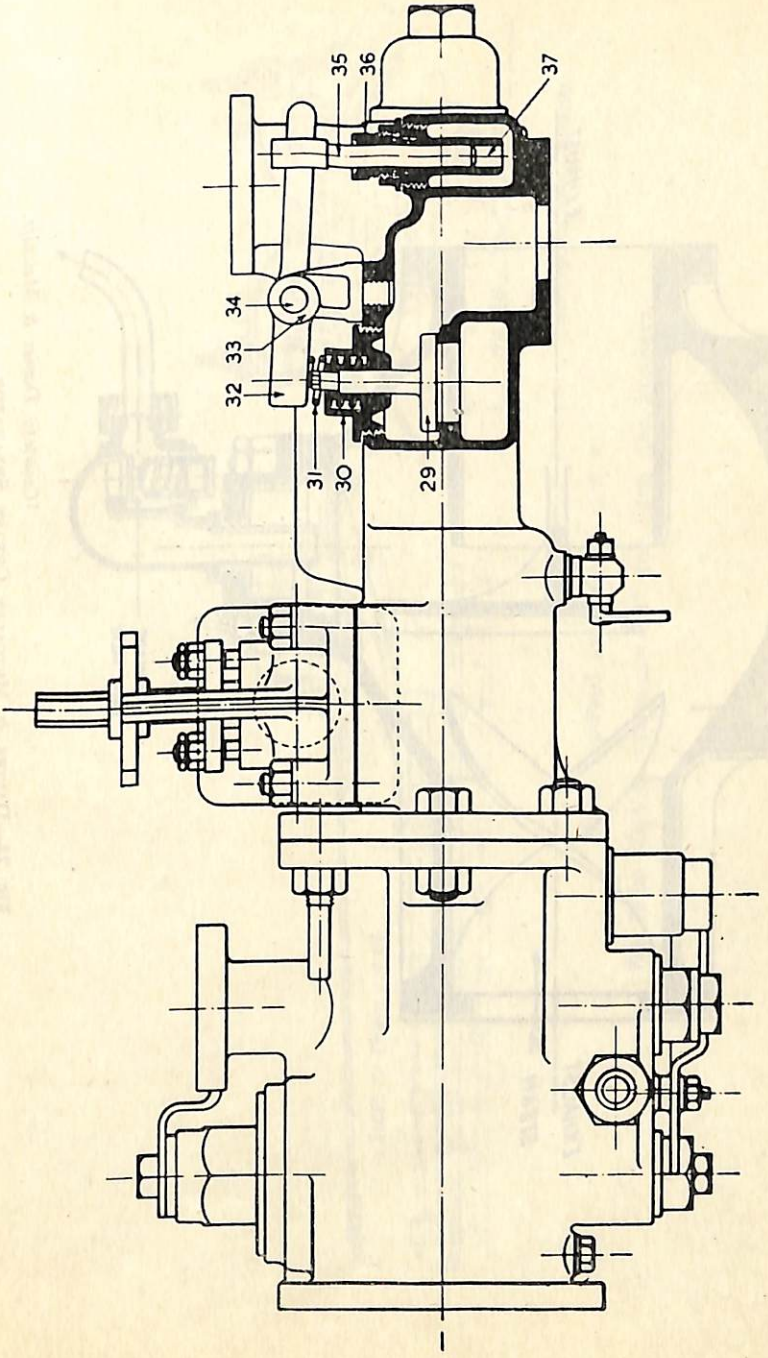
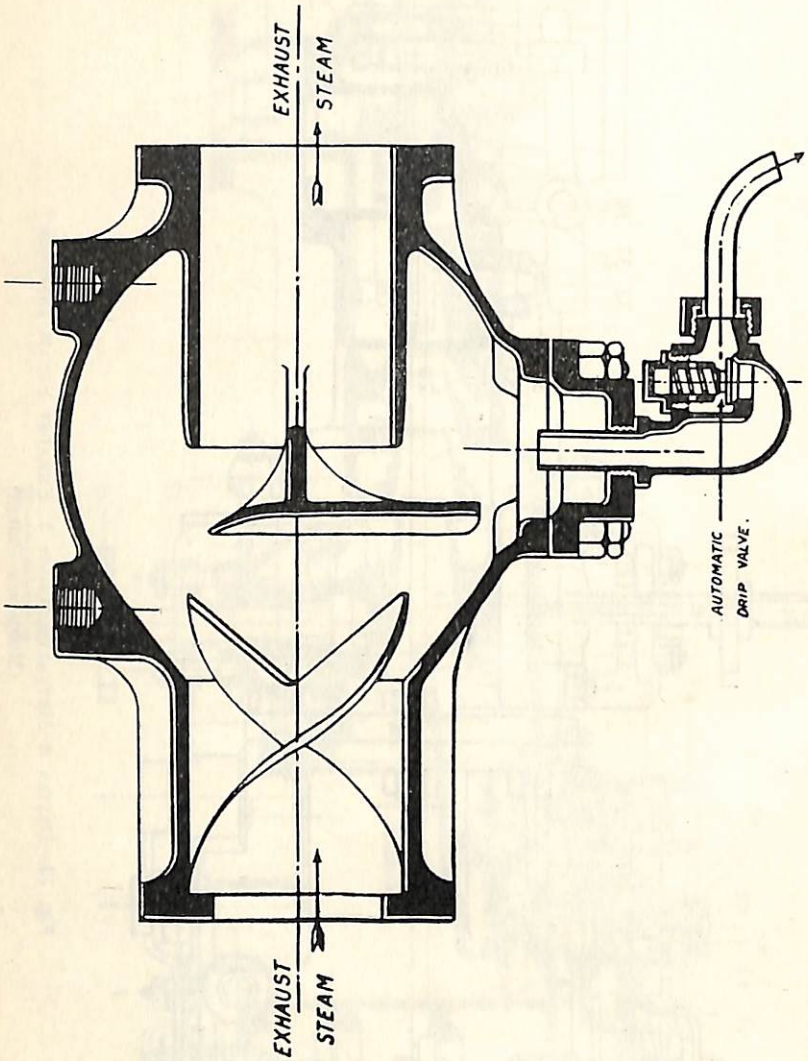
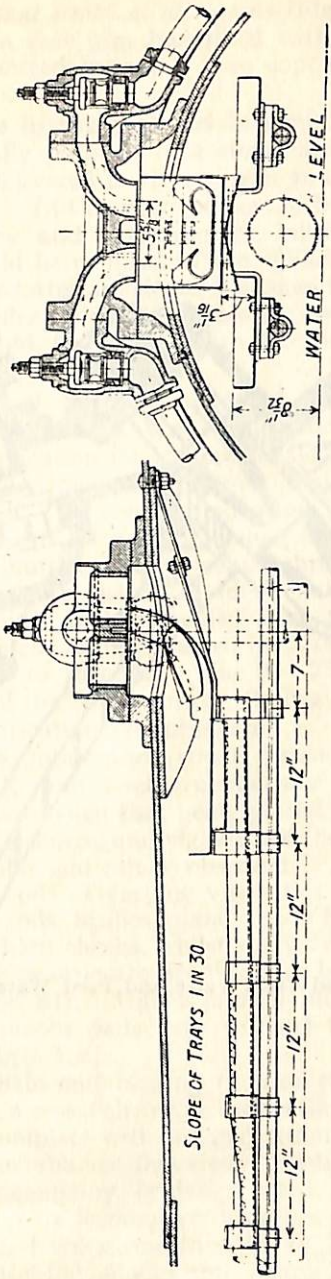


Fig. 73—Davies & Metcalfe, Class J. Exhaust Steam Injector. (Metcalfe's Patent).



[*Courtesy Davies & Metcalfe.*

Fig. 74—Davies & Metcalfe Grease Separator.



Boiler Top-Feed Arrangement, G.W.R.

[*Courtesy Railway Gazette.*

Fig. 75 (a)—G.W.R. "Churchward" Top Feed Arrangement.

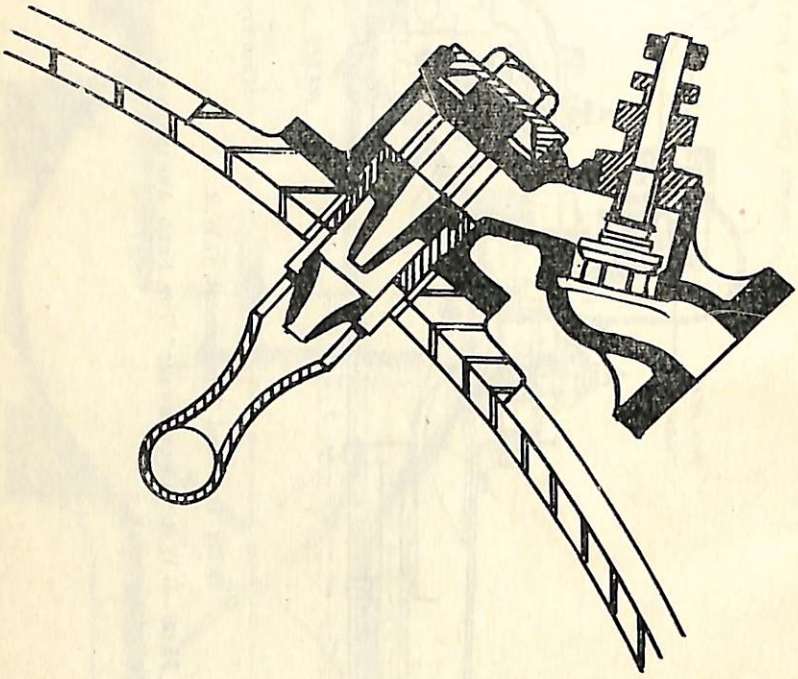


Fig. 75 (b)—Combined Clack Valve and Feed Water Heater.

having a constant stroke at all speeds (Fig. 79), operates a number of pumps in a cast iron box filled with lubricant. Each pump outlet is connected by small bore copper piping to the point of delivery.

Oil for use in the cylinders in conjunction with super-heated steam is usually pumped to a steam atomiser or anti-carboniser, whence it is delivered to the cylinder in a fine globular suspension by the steam. In this way the best of both the methods of mechanical delivery and displacement lubrication is sought. The atomiser should be coupled to the cylinder drain cock gear, to be open when the latter are closed (*i.e.*, engine working) and *vice versa*. Further, cylinder oil requires warming by a steam coil before it is sufficiently fluid to enable it to be pumped by the lubricator (Fig. 21*b*).

A number of different types of mechanical lubricators are available, those of C. C. Wakefield & Co., Ltd., probably having the widest application for locomotive use, whilst the "Silvertown" Lubricator and Atomiser was almost universally preferred by the former L.M.S.R. A recent innovation in this line has been the Davies & Metcalfe "F.S.A. Valveless Mechanical Lubricator" for cylinder and motion, and axlebox lubrication, which when used in conjunction with the oil divider, enables a maximum number of delivery points to be supplied from the minimum number of lubricators, which can be obtained in 12 or 24 feeds, with capacities ranging from 6 to 16 pints. The 90 "25" class 4-8-4 Condensing Locomotives of the South African Railways are fitted with this last mentioned lubrication (Frontispiece).

Hydrostatic lubricators (steam displaced) are still used widely (former G.W.R. and Southern Railway engines), one advantage claimed for these being that being placed in the cab they are at all times under the control and vigilance of the driver, and furthermore, the feed is visible and can be observed.

In general, oils of varying viscosities are fed to the cylinders, piston valves, rods, bushes, glands, slide bars, cross heads, axlebox bearings, and horn cheeks, whilst the valve gear components which to-day usually incorporate needle roller bearings, are grease pump lubricated (Fig. 81). Under journal lubrication is accomplished by spring emplaced pads with worsted trimmings carried in the sump or axlebox keep.

Big and little end bearing (non-roller) lubrication is effected by milling out a reservoir in the connecting rod end, and the fitting of a syphon complete with worsted trimming (Fig. 80) plunger, or externally adjustable needle valve, according to the general practice of the railway company, or design office.

For many years locomotive designers and operators, as well as civil engineers, have given thought to flange lubrication, in an effort to reduce flange and rail wear. The obvious danger of

lubricating the wheel rim must be guarded against (for reasons given in Chapter I). However, there is now available to locomotive designers the Davies & Metcalfe Automatic Wheel Flange Lubrication system which fulfils its object without incurring the danger referred to, oil being delivered to the wheel flanges from a mechanical lubricator by means of fine bore nozzles accurately directioned, it being claimed that prolonged tyre mileage by up to 300% has been obtained. As a practical example of its application, 18 class 10C. Beyer-Garratt locomotives for the Benguela Railway (Africa) are so fitted.

Finally, as previously, full information on lubricators and associated systems will be found in the bibliography, and upon request from the manufacturers of lubricating equipment.

(5) Steam Sanding, Sand Guns and Sootblowers.

The application of dry sand (in the moist temperate zones of the world) right under the driving wheel upon starting, particularly under adverse rail conditions (*e.g.*, dampness, greasiness, slime due to decomposed leaves alongside woods and forests), is a well known necessity. Gravity sanding, though still widely employed is not to be recommended, a more positive system employing steam (or compressed air) is advocated.

Essentially the apparatus (Fig. 83*a*) consists of an adequate reservoir of sand piped to within a few inches of the wheel at which it is to be delivered. A steam ejector is then screwed into the sand line, which upon the opening of a steam valve exhausts the sand pipe of air, and then blows the sand flowing down the sand pipe right under the driving wheel. A sand trap is usually placed at the outlet to the sand box which allows of the accumulation of sand but prevents its escape until it is drawn out by the evacuated air.

In an endeavour to keep the sand dry at all times within the sand box, British Railways standard locomotive types have steam heating coils (Downes Equipment) placed inside the sand boxes.

For locomotives working on very wind-exposed sections of track, there exists the Lambert Wet Sanding Apparatus whereby the sand is applied to the rail by a flush of water from the boiler (Fig. 83*b*).

For cleaning the fireside of the firebox tubeplate intermittently (about every 150 miles), the "Sand Gun" employing a stream of sand, or the "Soot Blower," a jet of steam, should be employed. With the former a small sand hopper is carried in the cab, sufficient for three applications. A steam jet carrying the sand along the inner firebox, is rotated by a hand-wheel, the sand nozzle being so shaped and positioned that the entire tubeplate is sprayed, thus clearing the tube ends of soot. With the soot blower, steam alone is employed. Messrs. Clyde Blowers, Ltd., Clydebank, and The

Superheater Co., Ltd., both manufacture sand guns, whilst the former are also famous for their wide range of soot blowers (Fig. 82).

(6) Further Locomotive Fittings.

Further locomotive fittings which are either obligatory or desirable include :—

(i) Water gauges (preferably two).
(ii) Whistle or whistles of distinctive note.
(iii) Fusible plugs, to be screwed into the inner firebox crown.
(iv) Carriage warming reducing valve and pressure gauge.
(v) Boiler and steamchest pressure gauges and pyrometer.
(vi) Speed indicators.
(vii) Electric lighting (steam turbo-generator) equipment for engine headlamp or lamps, inspection and oiling lights, tender lights, etc.
(viii) Automatic stoker (for grates in excess of 50 sq. ft.).
(ix) "Ajax" steam or air operated fire door.
(x) Power reverse gear.
(xi) Rocking grates and hopper ash pans.
(xii) Cylinder drain cocks and operating gear.
(xiii) Seating accommodation for driver and fireman.
(xiv) Look-out spectacles on the cab sides.
(xv) Steam manifold from which steam for all auxiliaries is taken.

All of these need to be investigated on their merits and where adopted the most suitable existing pattern should be decided upon with regard to the exigencies of locomotive design and the nature of the duties the locomotive is being designed to undertake. There is no guiding principle save that the final locomotive should be as trouble free, dependable, and easy to maintain and operate as possible ; for largely a locomotive's ultimate success depends upon its reputation with the staff whose duty it is to drive it and maintain it.

CHAPTER VII.

THE TENDER.

No survey of the steam locomotive could justifiably omit mention of that faithful hand maiden of all steam locomotives (other than tank engines and Garratts) the *Tender* (Fig. 84).

Considerable thought must be given to the design of the tender. Though its functions are few and simple it has always to play the

part of the intermediary between the locomotive proper and the train. The main function of a tender is to carry adequate (within reason) supplies of fuel and water for the locomotive, bearing in mind the availability of each on the one hand, and the desirability of as infrequent refuelling as possible (to make for increased engine availability) on the other. Its further function is to transmit the pull of the engine as smoothly as possible to the train, and to provide the train with a solid buffer against which to press either when running down hill or upon braking.

If common bituminous coal is the fuel to be carried the following points should assist in design :—

(i) One ton occupies approximately 45 cu. ft. (43-48 actual).

(ii) The coal bunker should be made self-trimming, *i.e.*, floor plate and side plates should slope downwards hopper fashion.

(iii) The bunker should be entered from the foot plate through a hinged door, which should extend to within 1 ft. of the shovelling platform when closed, and should be tightly closed when shut.

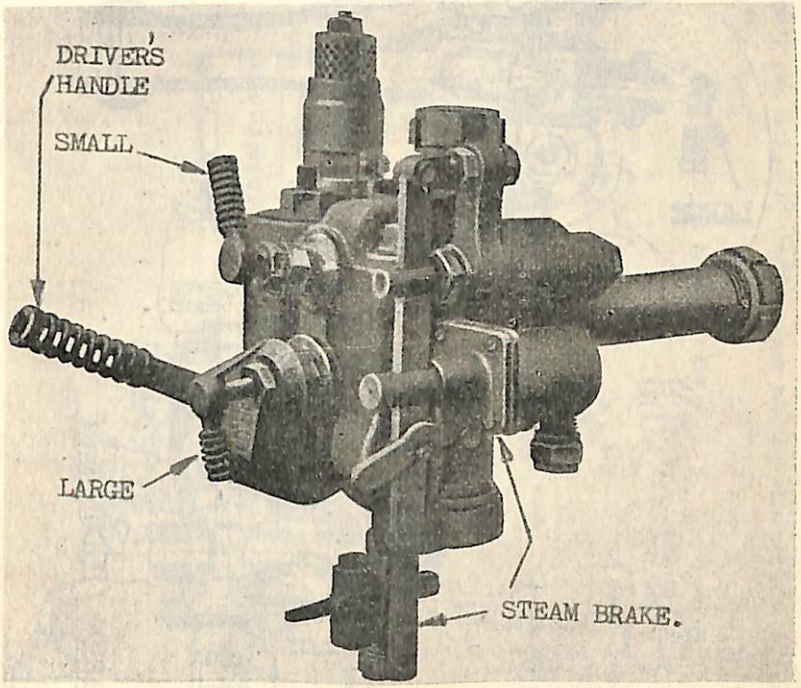
(iv) The shovelling platform should be about 1 ft. 6 in. above the cab floor, and there should not be a great disparity in elevation between it and the fire-hole door. Furthermore, if possible, it should project as far as possible into the cab, to reduce the distance between it and the fire-hole. The shovelling platform should be a little above the level of the fire hole, thus invoking the assistance of gravity when firing.

(v) Tender front plate grills should be forbidden, as they invite draughts, and encourage the blowing of coal dust into the cab.

(vi) The provision of a coal pusher (in the absence of an automatic stoker) though seldom provided, eliminates the dangerous need for the fireman or trimmer to mount the tender while the engine is in motion.

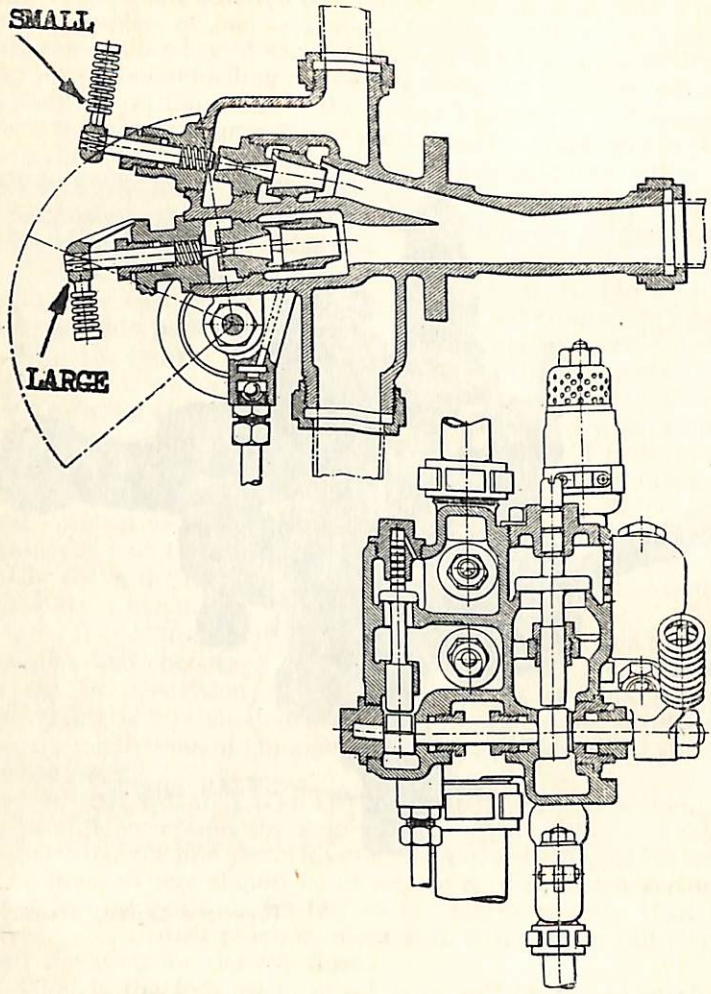
(vii) The cab foot plating should preferably be uniform, with the possible exception of a step to elevate the driver to enable him to see clearly the line ahead and to be out of the swing of the fireman as he fires. There should be at least 6 ft. 0 in. headroom for the latter who is more on his feet in his course of duty than is the driver. In British practice about 6 ft. 0 in. above rail level is a good elevation for the cab floor.

If oil is the fuel used, an oil tank will be placed in the coal space and provision must be made on the tender for all associated equipment usually carried thereon. The stoker engine, where an automatic stoker is employed, is also usually carried on the tender. It is also usual to carry vacuum or compressed air reservoirs on the tender, and accommodation must be found for firing irons, hand-tools, coal picks, flags, etc. Adequate locker space is essential, and the provision of a stainless steel lined, insulated food locker, as in latest British Railways' practice, should be obligatory.



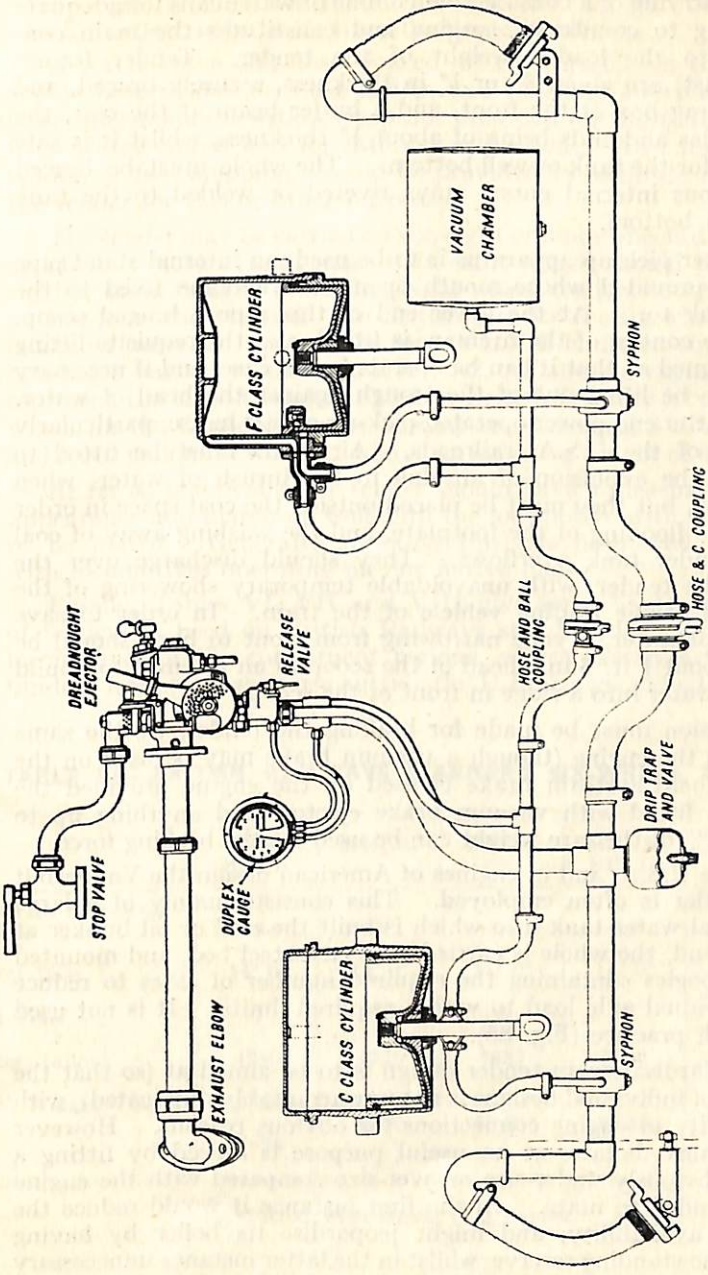
[*Courtesy Railway Gazette.*

Fig. 76 (a)—METCALFE'S VACUUM BRAKE EJECTOR WITH STEAM BRAKE VALVE.



[*Courtesy Railway Gazette.*

Fig. 76 (b)—Section Through Ejector.
Showing Nozzles—Steam, Exhaust and Train Pipes.
Bottom—Section Through Steam and Air Admission Valves.



[*Courtesy Railway Gazette.*

General Arrangement of Automatic Vacuum Brake Applied to Engine and Tender.

Fig. 77—Automatic Vacuum Brake Layout. Engine and Tender.

The carrying of a considerable volume of water calls for adequate reinforcing to counteract surging and constitutes the main contributor to the loaded weight of the tender. Tender frames (unless cast) are about $\frac{3}{4}$" or $\frac{7}{8}$" in thickness, securely braced, and carry a drag box at the front, and a buffer beam at the rear, the tender sides and ends being of about $\frac{1}{4}$" thickness, whilst it is safe to use $\frac{3}{8}$" for the tank or well bottom. The whole must be braced by generous internal gusset stays riveted or welded to the tank sides and bottom.

If water pick up apparatus is to be used, an internal stand pipe must be provided whose mouth opens into a dome fixed to the water tank top. At the lower end of this pipe a hinged scoop, under the control of the fireman, is fitted, and the requisite lifting gear designed so that it can be operated with ease, and if necessary the scoop be lifted out of the trough against the head of water. To this latter end power operated pick-up gear is in use, particularly on some of the U.S.A. railroads. Air vents must be fitted to facilitate the expulsion of air due to the inrush of water, when picking up, but they must be placed outside the coal space in order to obviate flooding of the footplate, and the washing away of coal if the tender tank overflows. They should discharge over the rear of the tender, with unavoidable temporary showering of the front end of the leading vehicle of the train. In order to save wastage of water, a vane narrowing from front to back should be placed about 1 ft. 6 in. ahead of the scoop in an attempt to "build up" the water into a ridge in front of the scoop.

Provision must be made for braking the tender, by the same means as the engine (though a vacuum brake may be used on the tender where a steam brake is used on the engine provided the engine is fitted with vacuum brake ejector) and anything up to 80% - 90% of the tare weight can be used for the braking force.

In the U.S.A. and in engines of American design the Vanderbilt type tender is often employed. This consists mainly of a large cylindrical water tank into which is built the coal or oil bunker at the cab end, the whole is carried on a cast steel bed, and mounted on two bogies containing the required number of axles to reduce the individual axle load to within required limits. It is not used in British practice (Fig. 85).

Standardisation in tender design is to be aimed at (so that the number of individual designs is not unwarrantably duplicated), with a similarity of engine connections for obvious reasons. However reason must dictate, as no useful purpose is served by fitting a tender obviously under-size or over-size compared with the engine it is intended to mate. In the first instance it would reduce the engine's availability, and might jeopardise its boiler by having little or no standing reserve, whilst in the latter instance unnecessary

dead weight would have to be hauled around by the locomotive, which in effect reduces its train hauling capabilities.

A water valve and hose worked off the injector on the engine should be provided for dust laying, and cab floor sweeping, and a large filler manhole with a clasp lid, provided for filling the tender from water cranes and for internal inspection. An external end ladder, or foot stepping, should be provided to facilitate mounting the tender for this or other purposes.

The tender may be carried on six, eight or more wheels depending on obvious factors ; not more than eight wheels should be housed rigidly, some designs preferring eight-wheel tenders to be carried on two four-wheel bogies. In British practice the six-wheel tender is almost universal, though the former L.N.E.R. fitted eight-wheel non-bogie tenders to their class A1, A2, A3 and A4 engines which represent the largest tenders fitted to British Railway locomotives, which (in the Corridor Tender design) accommodate 8 tons of coal and 5,000 gallons of water and have an inclusive weight of 62·4 tons.

As far as practicable the tender should have a pleasing external appearance and match up well with the locomotive with which it is to be mated. Generally speaking this has been well achieved with locomotives operating on our native lines.

Finally, Tables VI, VII and VIII give some broad indication of tenders as used on British Railways standard locomotives and locomotives of the constituent regions, whilst examples taken at random from engines in use outside the U.K. are given in Table IX.

TABLE VI.—BRITISH RAILWAYS STANDARD SIX-WHEEL TENDERS.

	BRI. BRI.B.	BRI.E.	BRI.F.	BRI.A. BRI.G.	BR.2.	BR.3.
Fitted to Std. Eng. Class	5-6-7	8	5-9	5-9	4-3	2
Wheel Base	14′ 0″	14′ 0″	14′ 0″	14′ 0″	13′ 0″	13′ 0″
Coal Tons	7	10*	7	7	6	4
Water Gallons	4250	4725	5625	5000	3500	3000
Total Weight Tons	49·20 50·25	55·50	55·25	52·50	42·15	36·85

*Fitted with steam coal pusher.

TABLE VII.—BRITISH RAILWAYS (Main Line Companies) TENDERS. Representative Selection.

EIGHT-WHEEL TENDERS.				
	L.N.E.R. "Corridor."*	L.N.E.R. "Non-Corridor."	S.R. Double Bogie.	M.O.S. W.D.
Coal—Tons, ...	8	9	5	9
Water—Gallons, ...	5,000	5,000	5,000	5,000
Wheel Diameter, ...	4′ 2″	4′ 2″	3′ 7″	3′ 2″
Wheelbase, ...	16′ 0″	16′ 0″	19′ 0″	15′ 9″
Total Weight, Tons, ...	62·4	60·35	57·55	55·5
Engine Type, ...	A.4. A.3.	A.1.-A.2. A.3.-A.4.	L.N. N.15.	2-8-0 2-10-0

* For non-stop runs, London (King's Cross)-Edinburgh—392·7 miles. "Flying Scotsman" (pre-war), and "The Capitals Ltd" and "The Elizabethan" (post-war). Engine crews changed en route, via the corridor.

TABLE VIII.—BRITISH RAILWAYS (Main Line Companies) TENDERS. Representative Selection—*continued.*

SIX-WHEEL TENDERS.									
	L.M.S. LARGE.	L.M.S. STD.	L.M.S. SMALL .	L.N.E.R. LARGE.	L.N.E.R. SMALL.	G.W.R. LARGE.	G.W.R. SMALL.	S.R. M.N., W.C.	S.R. STD.
Coal—Tons, ...	10†	9	4-6	7½	5½	6	5-6	5	5
Water—Gallons,	4,000	4,000	3,500	4,200	3,500	4,200	3,500	5,000-4,500	4.000
Wheel Diameter,	4′ 3″	4′ 3″	4′ 3″	3′ 9″	3′ 9″	4′ 1½″	4′ 1½″	3′ 7″	4′ 0″
Wheelbase, ...	15′ 0″	15′ 0″	13′ 0″	13′ 6″	13′ 0″	15′ 0″	15′ 0″	13′ 0″	13′ 0″
Total Weight, ...	56·35	54·65	42·7	51·0	44·2	46·7	40·0	50·0 42·6	42·4
Engine Type, ...	"Duchess"	Various	Various	J.39, V.2, B.1. K.3.	J.38, K.4. V.4.	"King" "Castle"	Various	"M.N." "W.C."	N.1. N.2. "Schools"

† Fitted with coal pusher.

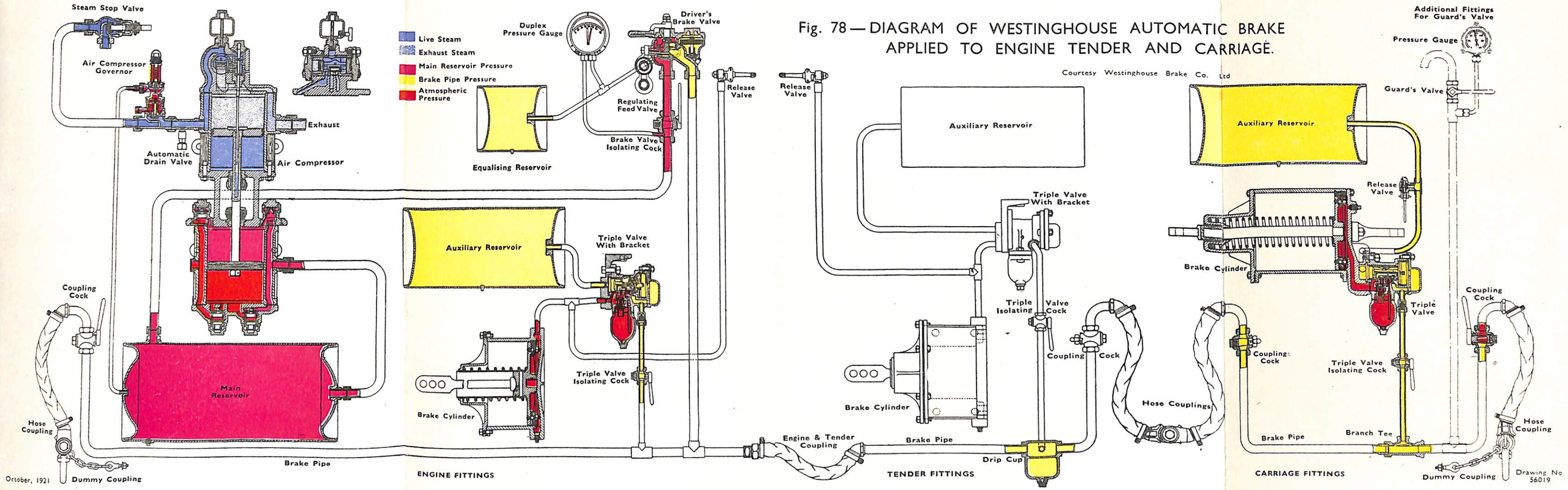
TABLE IX.—TENDER SIZES. NON-BRITISH.

	S.A.R. 25N.C.	S.A.R. 15 F.	I.G.R. W.G. & W.P.	Iranian State 2-10-2	Chinese Nat. 4-8-4	P.R.R. U.S.A. 6-8-6*	S.A.R. 24 & S.2†
Fuel	Coal	Coal	Coal	Oil	Coal	Coal	Coal
Fuel Tons/Gallons	18	14	18	1980	11·75	34	8
Water Gallons	10,500	6050	5000	6600	6600	18,000	4200
No. of Wheels	12	8	8	8	12	16	12
Wheel Base	32′ 0″	20′ 5″	22′ 6″	19′ 11″	29′ 8″	39′ 0″	24′ 7¼″
Total Weight Tons	108	68·1	71·16	67·27	76·9	184·3	54

*Pensylvania Rail Road 6-8-6 Steam Turbine.
†South African Railways Vanderbilt Type Tender.

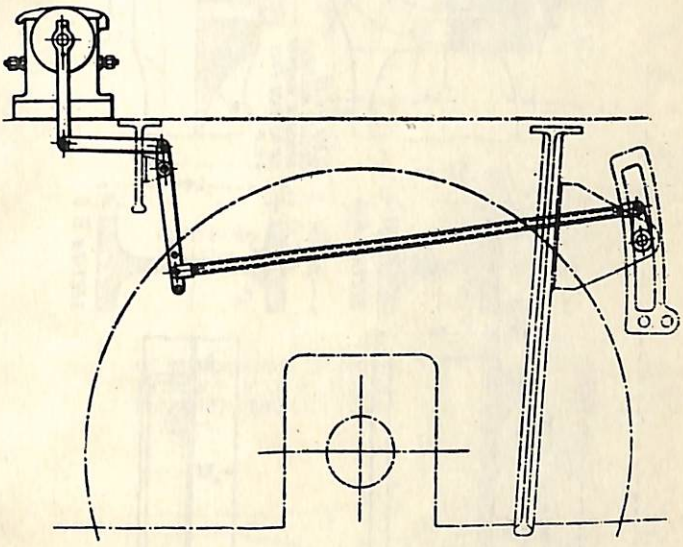
Fig. 78 — DIAGRAM OF WESTINGHOUSE AUTOMATIC BRAKE APPLIED TO ENGINE TENDER AND CARRIAGE.

Courtesy Westinghouse Brake Co. Ltd



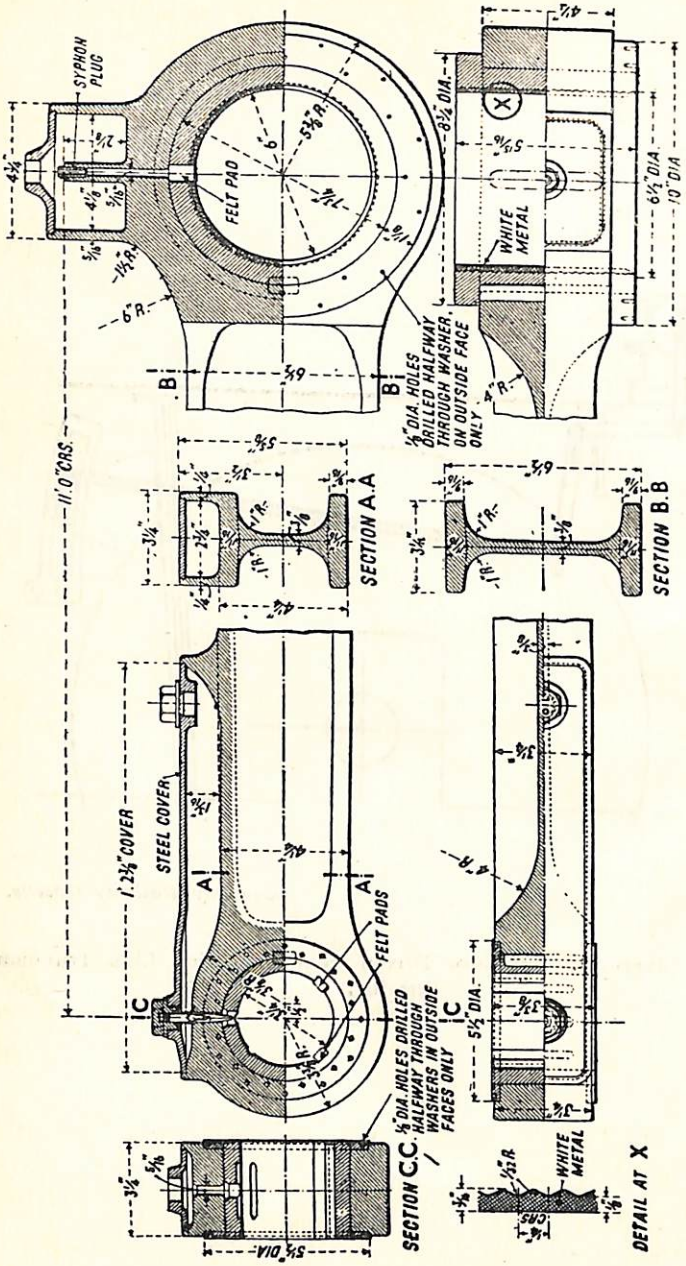
October, 1921

Drawing No 56019



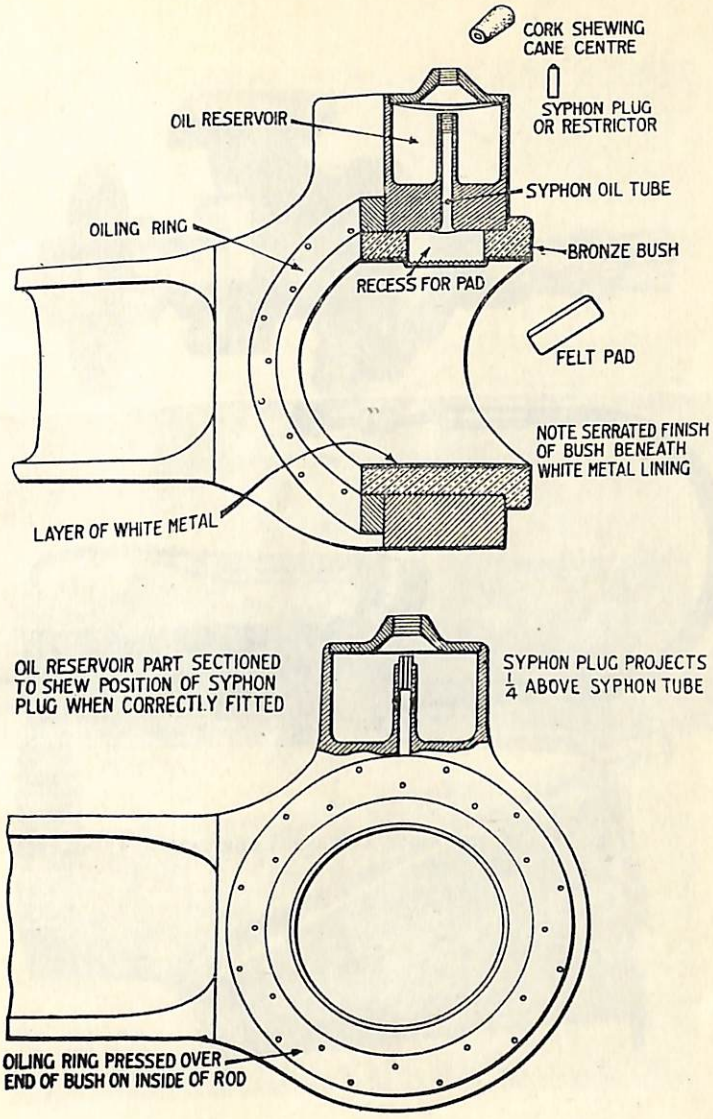
[*Courtesy Railway Gazette.*

Fig. 79—Silvertown Lubricator Driven from Expansion Link Trunnion Bearing.



[*Courtesy Railway Gazette.*

Fig. 80 (a)—Outside Connecting Rod of Four-Cylinder Express Engine, 4-6-2 Type, L.M.S.R.


CORK SHEWING CANE CENTRE
SYPHON PLUG OR RESTRICTOR
OIL RESERVOIR
SYPHON OIL TUBE
OILING RING
BRONZE BUSH
RECESS FOR PAD
FELT PAD
NOTE SERRATED FINISH OF BUSH BENEATH WHITE METAL LINING
LAYER OF WHITE METAL
OIL RESERVOIR PART SECTIONED TO SHEW POSITION OF SYPHON PLUG WHEN CORRECTLY FITTED
SYPHON PLUG PROJECTS 1/4 ABOVE SYPHON TUBE
OILING RING PRESSED OVER END OF BUSH ON INSIDE OF ROD

[*Courtesy Railway Gazette.*

Big-End Bearing—Plain Bush Type.

Fig. 80 (b)—Outside Connecting Rod and Big-End Lubrication.

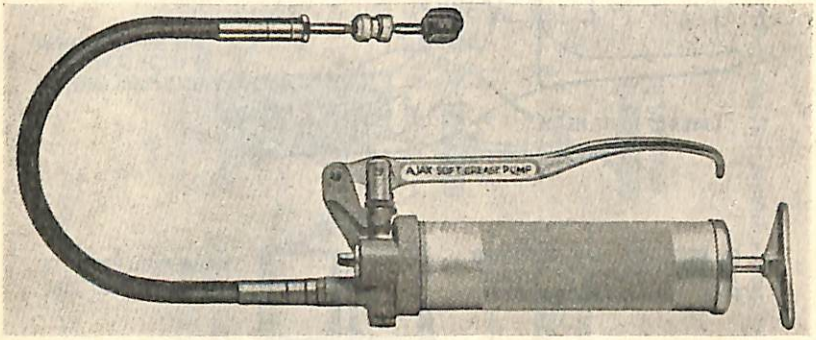
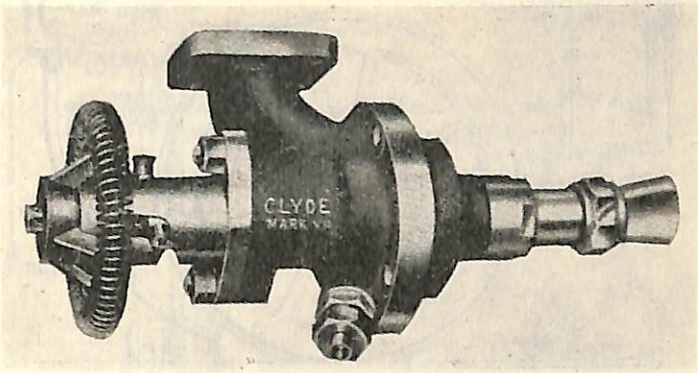
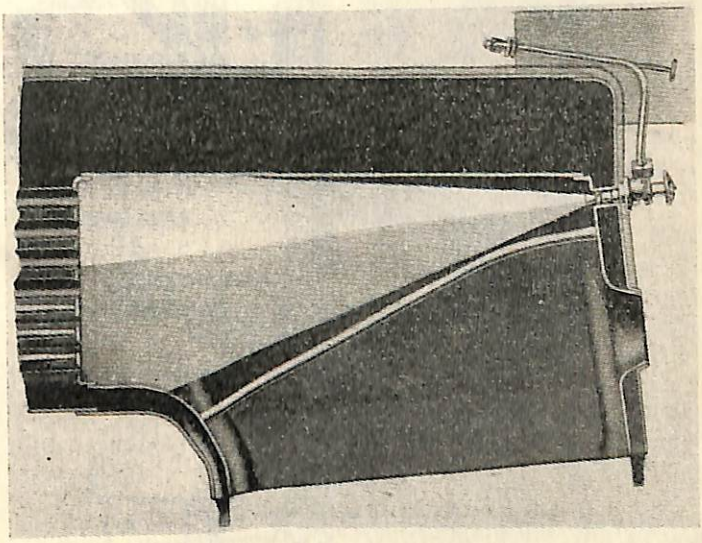


Fig. 81—"Ajax" Soft Grease Lubrication Pump.



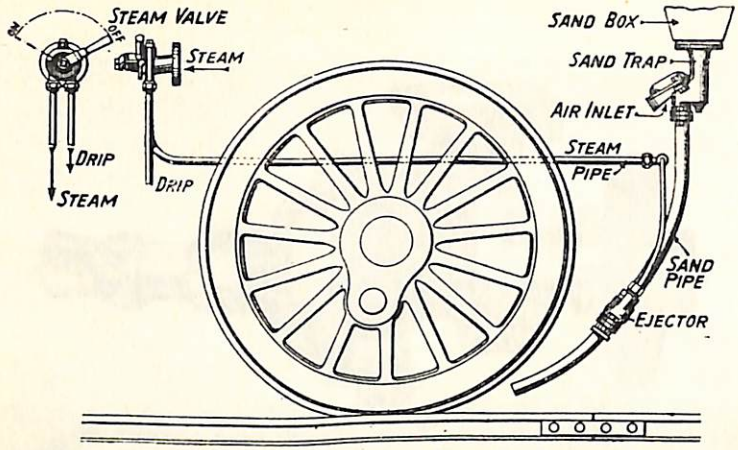
Clyde Back Plate Type Soot Blower.



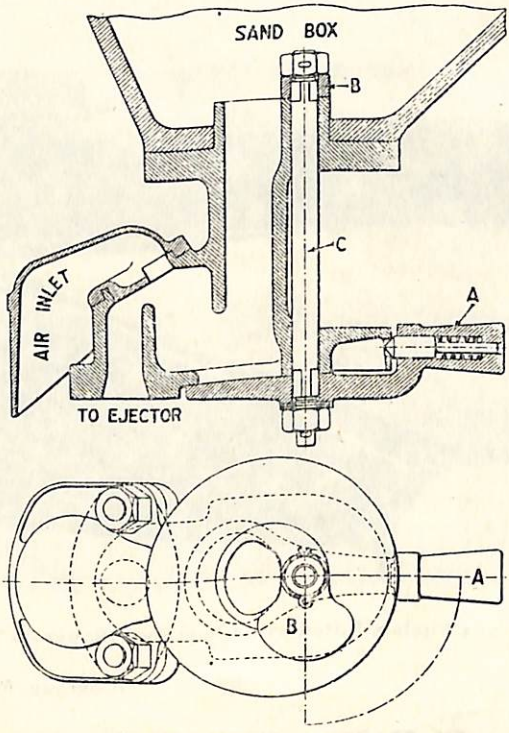
Sectional View of a Firebox Fitted with Clyde Soot Blower on the Back Plate.

[*Courtesy Railway Gazette.*

Fig. 82—The "Clyde" Soot Blower.

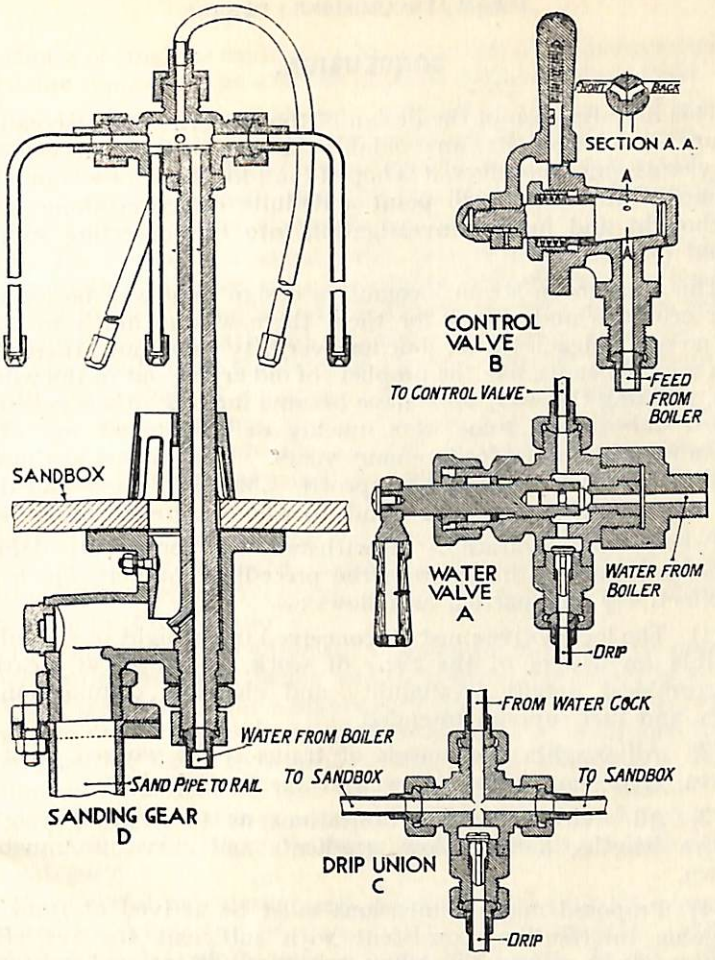


Arrangement of Steam Sanding Gear for Locomotives.

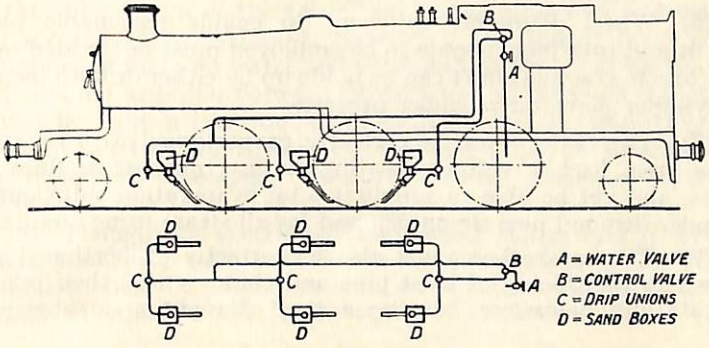


Section and Plan of Sandbox and Attachments.

Fig. 83 (a). [*Courtesy Railway Gazette.*



Details of Lambert Wet Sanding Gear.



Arrangement of Lambert Wet Sanding Gear on Locomotive.

Fig. 83 (b). [*Courtesy Railway Gazette.*

CONCLUSION.

This investigation of the design of the modern steam locomotive is now concluded, its unavoidable brevity having necessitated many omissions ; however, it is hoped that, for those for whom it was primarily intended it will point a definite direction along which all thought and further investigation into this diverting subject, should proceed.

The pioneers in steam locomotive design may well be forgiven their crudities and errors, for them there was so much to learn, and no one to teach them ; but for over fifty years now, there have been men of genius, like the prophets of old crying out in the wilderness, pointing the way, some have become famous, others remained almost unheard of, some were quickly to be listened to, whilst others went unheeded for too many years. Such names as Churchward, Walschaert, Schmidt, Caprotti, Chapélon, Goss and Fry, and others, spring readily to mind, to whom a great deal is owed.

Brevity and limitations notwithstanding, some very definite conclusions can be drawn from the preceding chapters which can be effectively summarised as follows :—

(1) The locomotive must be conceived in the light of the fullest possible knowledge of the type of work, geographical location, topographical details, availability and chemical composition of water and fuel supplies intended.

(2) All weights and speeds of trains to be worked must be known, giving maximum values and average speeds.

(3) All civil engineering limitations as to heights, weights, widths, lengths, hammerblow, gradients and curvature must be known.

(4) Proposed major dimensions must be arrived at from the foregoing information, consistent with sufficient tractive effort to overcome (i) gravity, (ii) rolling resistance, (iii) internal resistance, (iv) windage, and to accelerate the train to the required speeds.

(5) Wheel diameters sufficient to enable reasonable piston speeds and rotational speeds to be employed must be decided upon. Any loss in tractive effort can be made up by either or both increase in cylinder diameter or boiler pressure.

(6) The boiler must be correctly proportioned with regard to grate area, firebox volume, heating surface, number of flues and tubes, and yet be able to supply a total evaporation sufficient for cylinder demand plus steam demand for all steam using auxiliaries.

(7) The smoke box must also be correctly proportioned with regard to positioning of blast pipe and chimney and their relative heights and diameters, to give a good draughting arrangement.

The choice of single or multiple exhaust system must be investigated whilst the smoke box as a whole must be maintained air tight.

(8) All locomotives almost without exception, should be superheated. The gains greatly outweigh increase in costs and weights; inherent conservatism should be overcome and the likely advantages gained from the use of a steam dryer and or multiple valve regulator should be assessed.

(9) The choice of a sound valve gear is imperative, to enable the locomotive to boast a high cylinder performance, on the one hand (up to 80% being attainable by the use of a modern rotary cam poppet valve gear) and to enable it to slog away for long periods on a fully open regulator and a short "cut-off", on the other.

(10) Attention to detail design of motion components, axle boxes and their bearings, particularly in the metallurgical field, with a view to weight reduction and limitation of friction, is essential and beneficial, the use of roller and ball bearings wherever possible should be made.

(11) The locomotive must be carefully balanced with a view to compromising between hammerblow and swaying couple. Multi-cylinder locomotives can, if carefully designed, dispense with reciprocating balance.

(12) There must be sufficient thought given to a great number of accessories, some of which are obligatory mechanically or by statutory regulation, others may well be desirable, if not imperative, by nature of the duties to be performed and the conditions under which they are likely to occur. Such fittings as Injectors, Brake Equipment, Lubricators and Lubrication Systems, Sand Guns, and Soot Blowers, Turbo-generators, Sanding Gear, Rocking Grates, Hopper Ash Pans, Safety Valves and Cab Fittings are included in this category.

(13) Careful cab design and disposition of controls normally operated by driver and fireman, duplication of regulator, whistle, injector control (and in some schools of thought, brake handle) being very desirable.

(14) The design of the tender (or bunkers for fuel and water) must incorporate considerable thought. The tender must carry all supplies of fuel and water and sundries (except sand) on the one hand, and act as buffer between locomotive and train, on the other. It must be rigid in itself but flexible on the track; unfortunately its weight must be regarded as dead weight which is incapable of being effectively employed. It must, however, possess good riding qualities.

(15) Though not specifically mentioned in the text it will be realised that with tank engines the water tanks on either side of the boiler must be interconnected in order to maintain lateral equilibrium, and the side tanks must be connected to the water

tank under the fuel space, if there is one, to help distribute the weight of water. A common feature of tank locomotives and "Garratts" is that their total weight diminishes and hence their available adhesion, as the fuel and water are consumed. This is not the case with a tender locomotive (excluding the tender) whose adhesion remains constant.

(16) Thus in the light of all known and foreseeable conditions the locomotive should be designed so that it will be :—

(*a*) Easy to construct in the first instance.

(*b*) Easy and pleasing to operate.

(*c*) Easy to maintain from a point of view of accessibility.

(*d*) Economical to maintain and operate, with the greatest possible degree of mechanical reliability, and as low a fuel, water, and oil (lubricating) consumption as possible consistent with efficiency and a good power output.

ACKNOWLEDGMENTS.

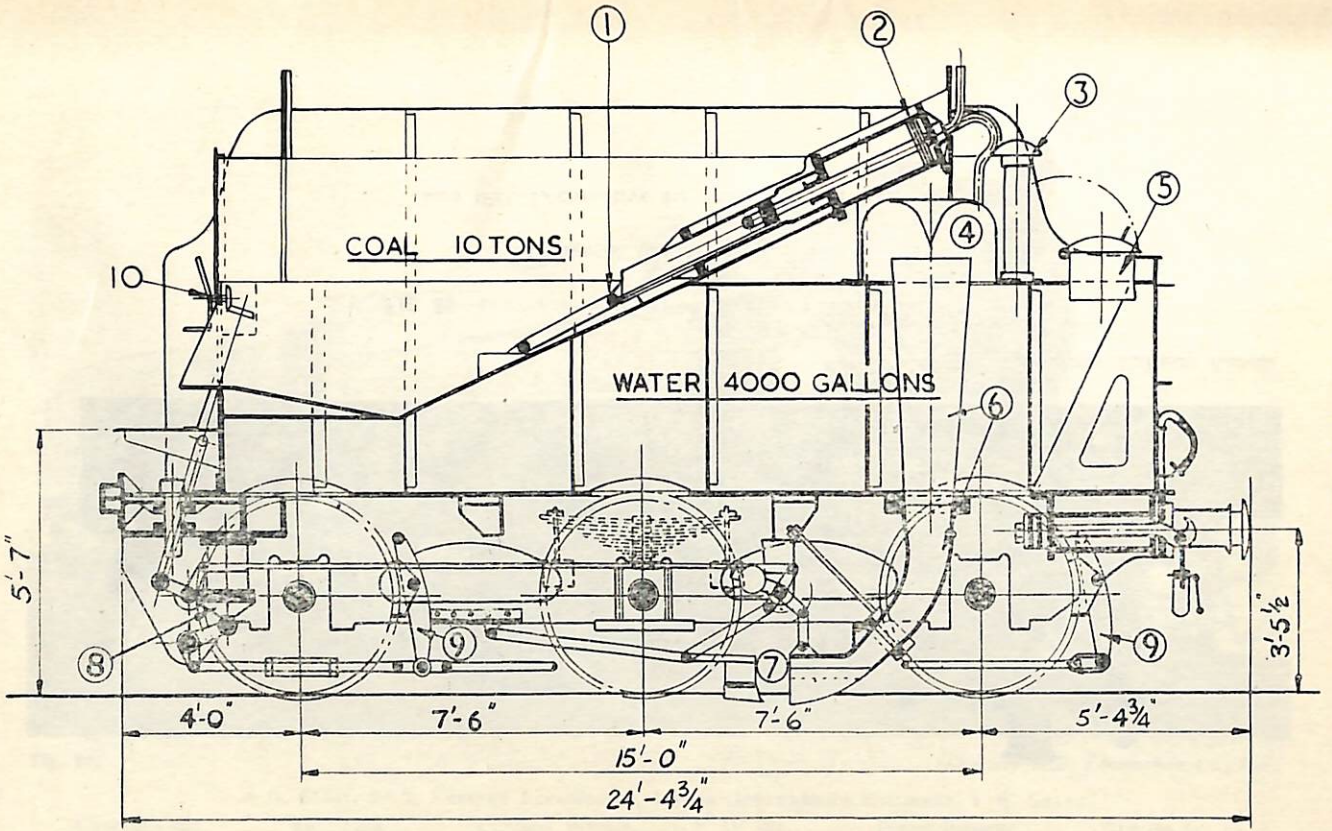
There remains the pleasurable duty of gratefully acknowledging all those firms of manufacturers of locomotive equipment for their assistance and permission to use material readily supplied in response to the author's request.

Acknowledgments and thanks are due to :—

Beyer-Peacock & Co., Ltd. "Beyer-Garratt" Locomotives.
Associated Locomotive Equipment Ltd. ... Rotary Cam Poppet Valve Gears.
The Vacuum Brake Co., Ltd.
The Westinghouse Brake and Signal Co., Ltd. ... ... } Vacuum and Compressed Air Brake Systems.
Davies & Metcalfe, Ltd. ... { Vacuum Brake Ejectors. Valveless Lubricators. Wheel Flange Lubricators. Injectors.
Whitelegg & Rogers, Ltd., ... "Ajax" Firedoor and Grease Pump.
Gresham & Craven, Ltd. ... ... Injectors.
The Superheater Co., Ltd. ... ... Superheaters and Dryers.
R. L. Ross & Co., Ltd. (Stockport) ... ... Safety Valves.
"The Railway Gazette"
North British Locomotive Co., Ltd.

The Locomotive Publishing Co., Ltd., and The Caxton Press, without whom much material and illustrative matter could not have been presented, and British Railways for the use of official photographs.

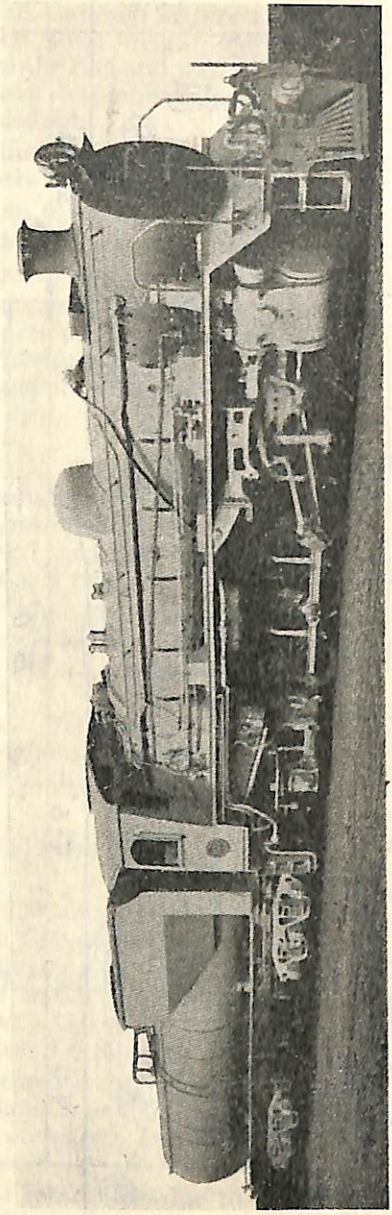
Personal thanks are due to Messrs. W. Stewart, L. T. Daniels, P. T. Creedy, J. S. Cowan and A. J. Bannister for their valuable assistance.



KEY.

1. Steam Operated Coal Pusher.
2. Coal Pusher Piston.
3. Air Vent Pipe.
4. Water Pick Up Dome.
5. Water Filler Manhole.
6. Water Pick Up Standpipe.
7. Pick Up Scoop and Vane.
8. Steam Brake Cylinder.
9. Break Gear.
10. Pick Up and Hand Brake Screw Operating Handle.

Fig. 84—L.M.S. 4,000 Gallon Large Tender. 8P. Class Locomotives. Simplified Section.



[*Courtesy Railway Gazette.*

Fig. 85—Locomotive With Vanderbilt Type Tender.

Gauge of Railway, 3 ft. 6 in.

Class 19D. Locomotive for South African Railways.

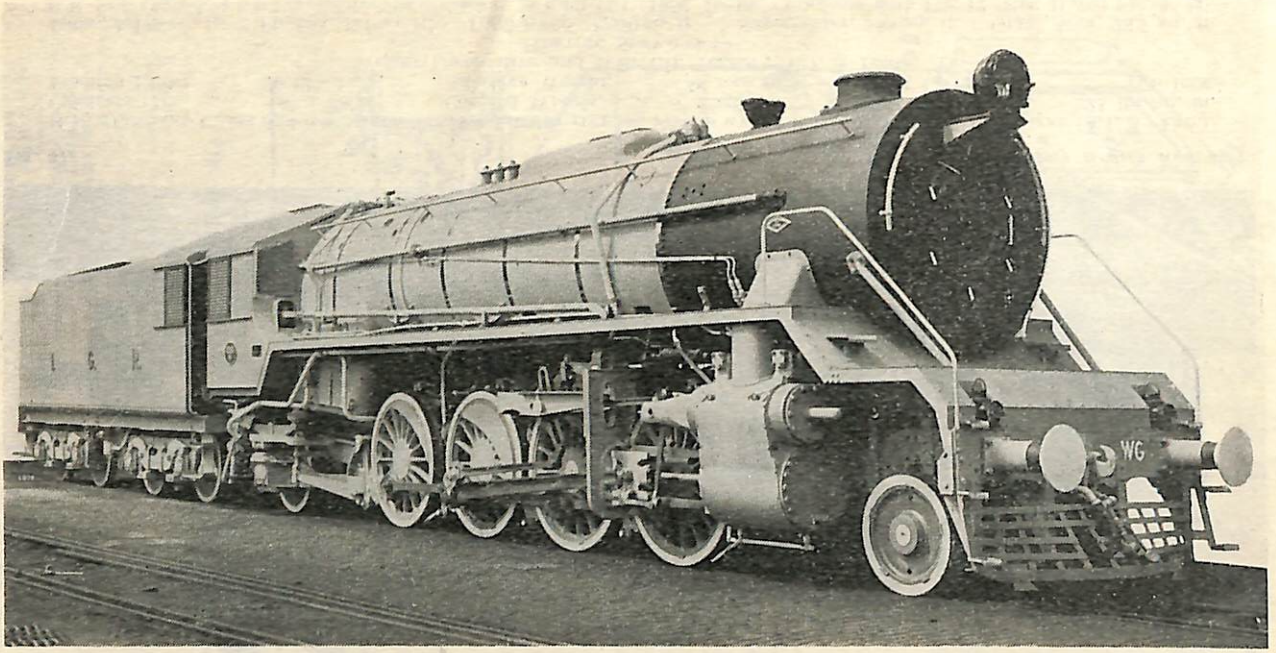


Fig. 86. [*Courtesy N.B. Locomotive Co., Ltd.*

W.G. Class, 2-8-2, Freight Locomotive, Indian Government Railways, 5′ 6″ Gauge.

Cylinders (2), ... 21⅞″ × 28″. Coupled Wheels, ... 5′ 1⅛″ dia. Boiler Pressure, ... 210 lbs./sq. in.
Tractive Effort, ... 38,890 lbs. Weight Engine, ... 101.85 tons. Weight Engine and Tender, 173.7 tons.
A locomotive of this Class was exhibited at the Festival of Britain, 1951, Exhibitions at South Bank (London), and St. Georges Square, Glasgow.

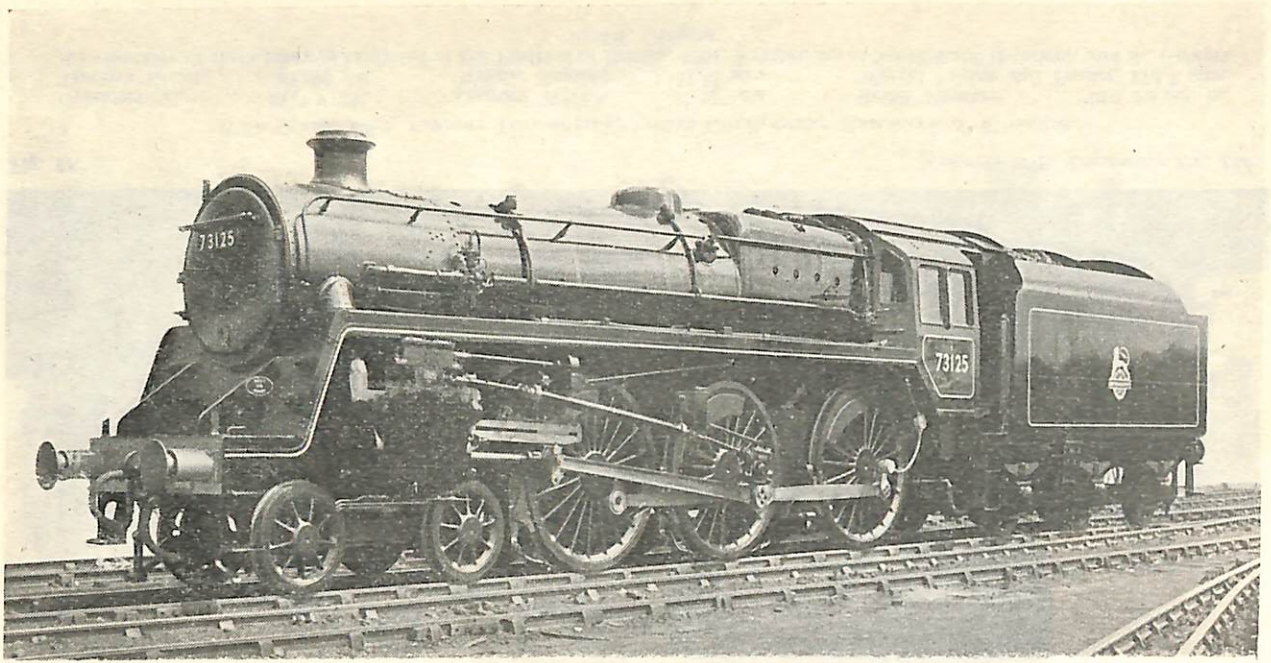
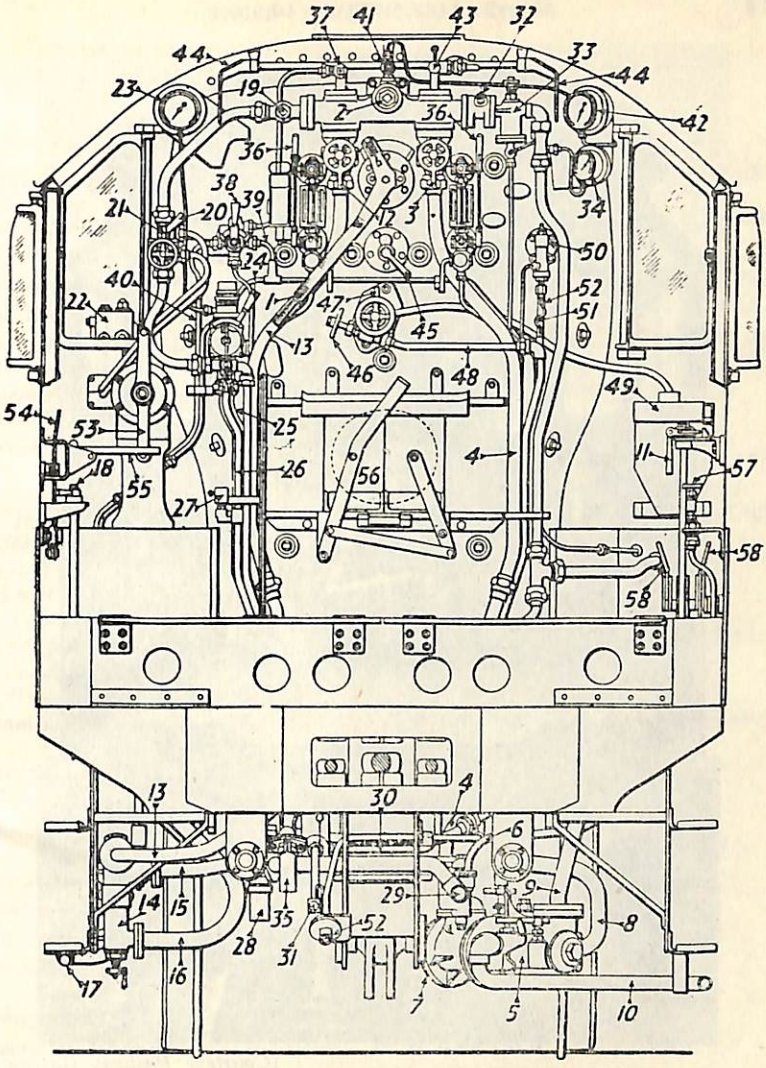


Fig. 87. [*Courtesy British Railways.*

B.R. Standard Class 5, 4-6-0, Mixed Traffic Locomotive, as Fitted with British-Caprotti Valve Gear (A.L.E., Ltd.).
Cylinders (2) ... 19″ × 28″. Coupled Wheels, ... 6′ 2″ dia. Boiler Pressure, ... 225 lbs./sq. in.
Tractive Effort, ... 26,120 lbs. Adhesive Weight, ... 58 tons. Weight Locomotive, ... 76 tons.
Weight Locomotive and B.R. 1.B. Tender : 127. 5 Tons.
Heating Surfaces :—
Tubes, 1,479 sq. ft. Firebox, 171 sq. ft. Total Evap., 1,650 sq. ft. Superheater, 358 sq. ft. Grate Area, 28.7 sq. ft.
B.R. Std. Class 5, 4-6-0 Locomotive, final version, as built Oct., 1956, Derby. Compare with Fig. 12 (Part 1) and Fig. 55 (*a*).

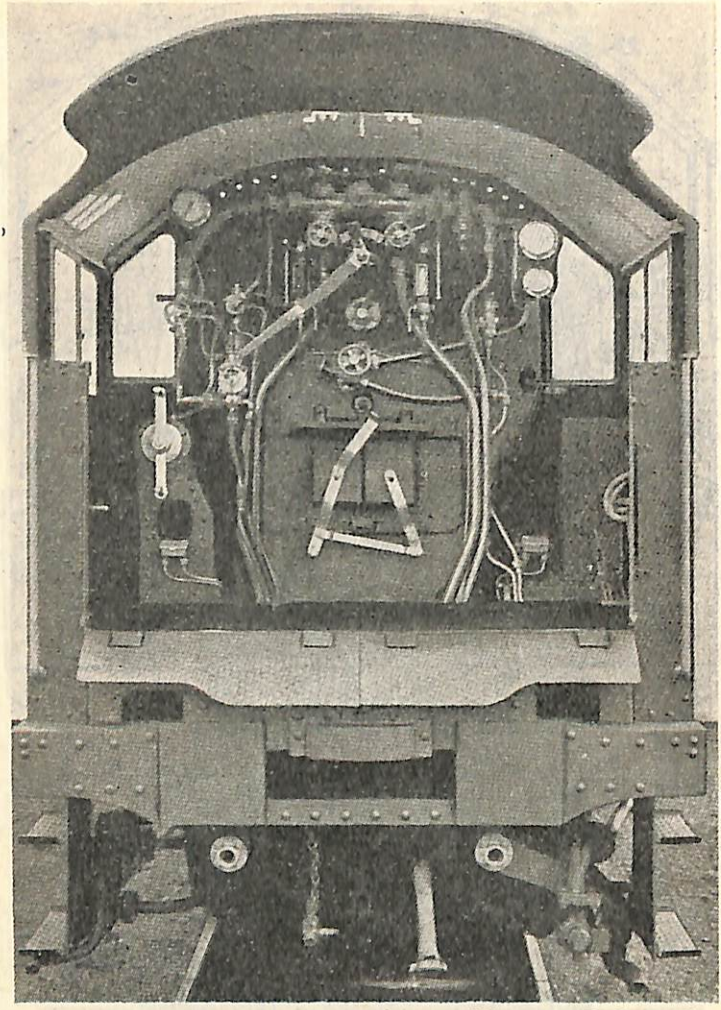


[*Courtesy Railway Gazette.*

Fig. 88 (a)—Arrangement of Footplate Fittings for L.M.S.R. Class 5 Locomotive.

1. Regulator handle.
2. Main steam valve for steam supply stand.
3. Live steam valve to exhaust steam injector.
4. Live steam pipe to injector.
5. Exhaust steam injector.
6. Auxiliary steam pipe from steamchest to injector.
7. Exhaust steam pipe to injector.
8. Water-feed pipe to injector.
9. Delivery pipe from injector to boiler.
10. Overflow pipe from injector.
11. Water-control gear for injector.
12. Steam valve to live steam injector.
13. Live steam pipe to injector.
14. Live steam injector.
15. Water-feed pipe to injector.
16. Delivery pipe from injector to boiler.
17. Overflow pipe from injector.
18. Water-control gear for injector.
19. Stop valve to ejector steam valves.
20. Small ejector steam valve.
21. Large ejector steam valve.
22. Combined large and small ejector.
23. Vacuum gauge.
24. Driver's brake valve.
25. Train pipe.
26. Steam brake pipe.
27. Steam brake cylinder lubricator.

(*Continued overleaf.*)



[*Courtesy Railway Gazette.*

Fig. 88 (b)—View of Foorplate. L.M.S. 6P., 4-6-0.

(*Continued.*)

28. Drip valve for train pipe.
29. Train pipe connection to tender.
30. Steam pipe to engine brake cylinder.
31. Steam brake pipe connection to tender.
32. Stop valve to carriage-warming reducing valve.
33. Carriage-warming reducing valve.
34. Carriage warming pressure gauge.
35. Carriage-warming hose-pipe connection to tender.
36. Water-gauge cocks.
37. Stop valve to sanding valve.
38. Steam sanding valve.
39. Sand pipe to trailing wheels.
40. Sand pipe to leading and driving wheels.
41. Steam valve for boiler-pressure gauge.
42. Boiler-pressure gauge.
43. Whistle valve.
44. Whistle handles.
45. Blower valve.
46. Steam valve for sand gun.
47. Sand gun hand-operating wheel.
48. Steam pipe from steamchest to sand gun.
49. Sand hopper for sand gun.
50. Continuous blow-down valve.
51. Steam pipe from steamchest to blow-down.
52. Blow-down pipe to tender.
53. Reversing screw handle.
54. Cylinder frain cock handle.
55. Driver's seat.
56. Firehole door.
57. Coal-watering cock.
58. Ashpan handles.

BIBLIOGRAPHY

(other than Periodicals referred to in the text).

Railways of To-day	Cecil J. Allen, M.Inst.T., A.I.Loco.E. (Frederick Warne & Co., Ltd).
Locomotive Practice and Performance (in the 20th Century)	Cecil J. Allen, M.Inst.T., A.I.Loco.E (Heffer & Son, Ltd.).
Railwaymens' Gallery	Roger Lloyd (George Allen & Unwin).
The Steam Locomotive To-day ...	M. P. Sells, O.B.E. (Loco. Pub. Co. Ltd.).
Steam Locomotive Design ... (Data and Formulae)	E. A. Phillipson, A.M.I.Mech.E., M.I.Loco.E. (Loco Pub. Co., Ltd.).
Locomotive Engineers Pocket Book (42nd Edition)	Loco. Pub. Co., Ltd.
Locomotives of Sir Nigel Gresley	O. S. Knock, B.Sc., M.I.Loco.E. (The Railway Publishing Co., Ltd.). (Longmans Green & Co., Ltd.).
A British Railway Behind the Scenes	J. Williamson, B.Sc. (Ernest Benn Ltd).
Pocket Book for Mechanical Engineers	D. A. Low, M.I.Mech.E. B. B. Low, B.A.
A Manual of Machine Drawing and Design	D. A. Low, M.I.Mech.E. both (Longmans Green & Co., Ltd.).
Locomotive Management ... ... (8th and 10th editions)	(Railway Publishing Co., Ltd.).
Steam Locomotion I. & II. ...	E. C. Poultney, O.B.E. (The Caxton Publishing Co., Ltd.).
The Metallurgy of a High Speed Locomotive	(The Railway Gazette, 1938).
Locomotive Valves and Valve Gears	C. S. Lake & A. Reidinger. (Percival Marshall & Co., Ltd.).
Locomotive Superheaters	The Superheater Co., Ltd.
Great Northern Locomotives ...	R. A. H. Weight.
British Locomotive Types ...	(Railway Publishing Co., Ltd.).
Locomotives of R. E. L. Maunsell	O. S. Knock, B.Sc., M.I.Loco.E. (Everrard).
Theory of Machines	Robert F. McKay (Arnold).
Theory of Machines	B. B. Low, B.A. (Longmans).
Theory of Machines	Bevan (Longmans).
Metallurgy for Engineers ...	Gregory (Blackie).
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20. Referred Mean Pressure Cut-Off, etc.
21. Particulars for Balata Belt Drives.
22. ⅞″ Square Duralumin Tubes as Struts.
23. 1″ ,, ,, ,,
24. ¾″ Sq. Steel Tubes as Struts (30 ton yield).
25. ⅞″ ,, ,, ,, (30 ton yield).
26. 1″ ,, ,, ,, (30 ton yield).
27. ¾″ ,, ,, ,, (40 ton yield).
28. ⅞″ ,, ,, ,, (40 ton yield).
29. 1″ ,, ,, ,, (40 ton yield).
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31. Moments of Inertia of Built-up Sections (Instructions and Examples). }
34. Capacity and Speed Chart for Troughed Band Conveyors.
35. Screw Propeller Design (Sheet 1, Diameter Chart). } Connected.
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37. ,, ,, ,, (Sheet 3, Notes and Examples). }
38. Open Coil Conical Springs.
39. Close Coil Conical Springs.
40. Trajectory Described by Belt Conveyors (Revised 1949).
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46. Coned Plate Development.
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48. Solution of Triangles (Sheet 2, Oblique Angles).
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55. ,, ,, ,, (Square Wire).
} Connected.
56. Relative Value of Welds to Rivets.
58. Graphs for Strength of Rectangular Flat Plates of Uniform Thickness.
59. ,, Deflection ,, ,, ,, ,, ,,
61. Deflection of Leaf Spring.
62. Strength of Leaf Spring.
63. Chart Showing Relationship of Various Hardness Tests.
64. Shaft Horse-Power and Proportions of Worm Gears.
65. Ring with Uniform Internal Load (Tangential Strain)
66. ,, ,, ,, ,, (Tangential Stress)
} Connected.
67. Hub Pressed on to Steel Shaft. (Maximum Tangential Stress at Bore of Hub).
68. Hub Pressed on to Steel Shaft. (Radial Gripping Pressure between Hub and Shaft).
69. Rotating Disc (Steel) Tangential Strain.
70. ,, ,, ,, Stress.
} Connected.
71. Ring with Uniform External Load, Tangential Strain.
72. ,, ,, ,, ,, ,, Stress.
} Connected.
73. Viscosity Temperature Chart for Converting Commercial to Absolute Viscosities.
74. Journal Friction on Bearings.
75. Ring Oil Bearings.
} Connected.
76. Shearing and Bearing Values for High Tensile Structural Steel Shop Rivets, in accordance with B.S.S. No. 548/1934.
78. Velocity of Flow in Pipes for a Given Delivery.
79. Delivery of Water in Pipes for a Given Head.
} Connected.
80. (See No. 105).
81. Involute Toothed Gearing Chart.
83. Variation of Suction Lift and Temperature for Centrifugal Pumps.
89. Curve Relating Natural Frequency and Deflection.
90. Vibration Transmissibility Curve for Elastic Suspension.
91. Instructions and Examples in the Use of Data Sheets, Nos. 89 and 90.
} Connected.
92. Pressure on Sides of Bunker.
93-4-5-6-7. Rolled Steel Sections.
98-9-100. Boiler Safety Valves.
102. Pressure Required for Blanking and Piercing.
103. Punch and Die Clearances for Blanking and Piercing.
104. Nomograph for Valley Angles of Hoppers and Chutes.
105. Permissible Working Stresses in Mild Steel Struts to B.S. 449, 1948.
106. Compound Cylinder (Similar Material) Radial Pressure of Common Diameter (D1).
107. True Angles for Pipe Bends.
108. Development of Spiral Chutes.

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